COMBUSTION

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DEVOTED TO THE ADVANCEMENT OF STEAM PLANT DESIGN AND OPERATION

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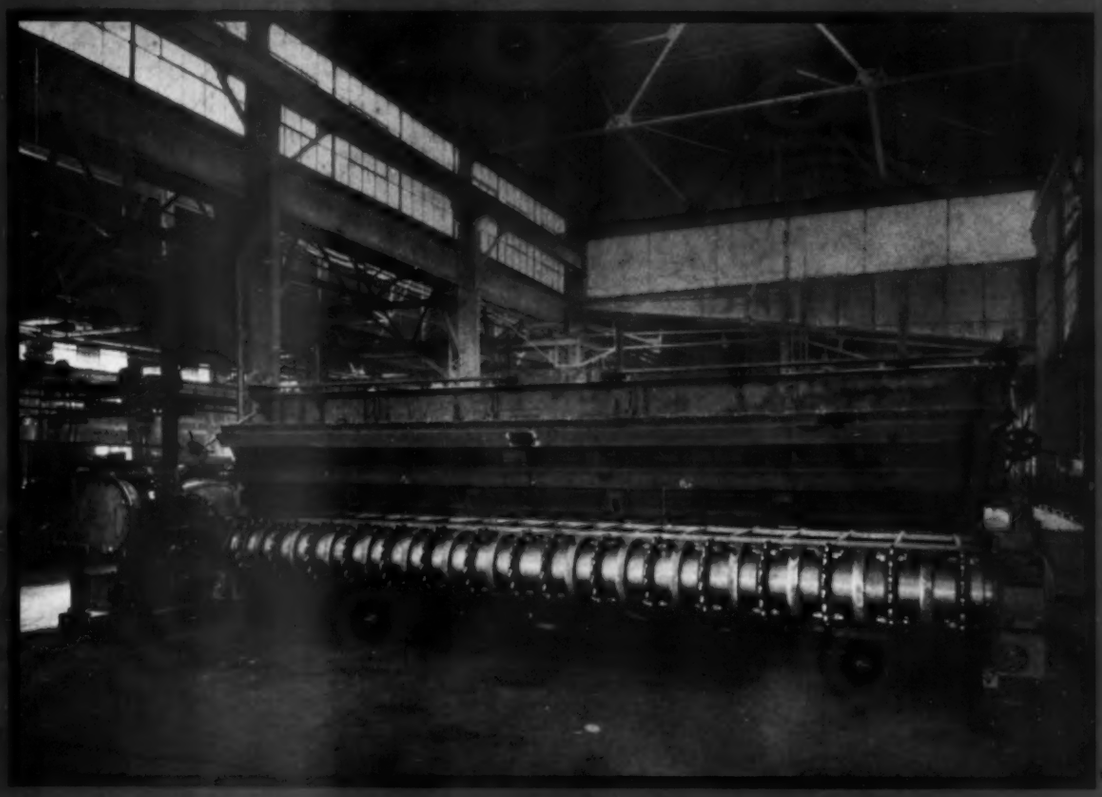
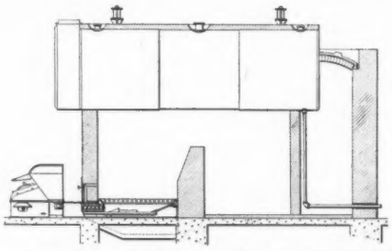


Shop view of large forced-draft, chain-grate stoker being installed under 1350-lb boiler at the Omaha Station of the Nebraska Power Company. This stoker is 25 ft. wide and 22 ft, 9 in. long, and is the largest stoker of this type yet built in this country

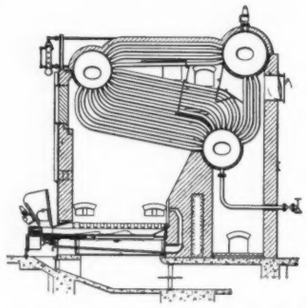
Some High Spots of the A.S.M.E. Annual Meeting

High-Pressure Installation at Mannheim

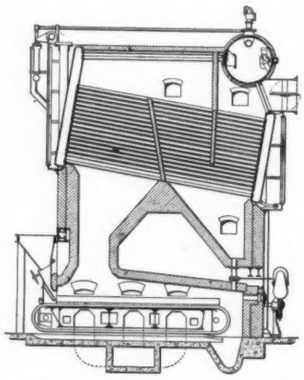
MODERN STEAM GENERATING UNITS for small and medium-sized plants



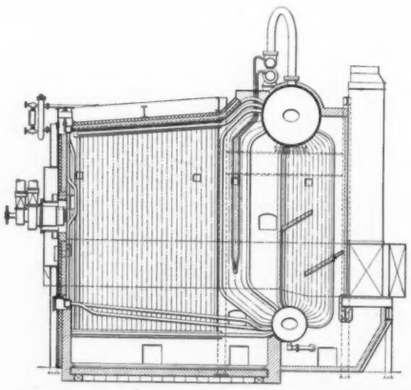
Unit comprised of CE Fire Tube Boiler and CE-Skelly Stoker Unit. Boiler h.s.—1525 sq ft Similar units are available in sizes down to 300 sq ft



Unit comprised of CE Bent Tube Boiler (Type VM) and CE Underfeed Stoker. Boiler h.s.—2250 sq ft



Unit comprised of CE-Heine Box Header Boiler and CE-Coxe Traveling Grate Stoker. Boiler h.s.—5000 sq ft



CE Steam Generator (Type VU). Maximum continuous output—75,000 lb per hr. Units of this type are available for capacities from 20,000 to several hundred thousand lb of steam per hr

The accompanying drawings of units installed or on order are representative of CE installations in small and medium-sized plants. Each of the various types of boilers and firing equipment is available in a range of sizes adequate for the conditions to which it is applicable. These products, in conjunction with others, comprising the complete CE line, enable Combustion Engineering to supply the most suitable and satisfactory types of equipment for any combination of load and fuel conditions.

Although CE installations include many of the largest and most notable steam generating units in service, they also include a great number of small and medium-sized units which are equally outstanding in their class. By virtue of the broad experience reflected in their design and application, such units give optimum economic results for the conditions under which they operate. They are a source of pride and profit to their owners.

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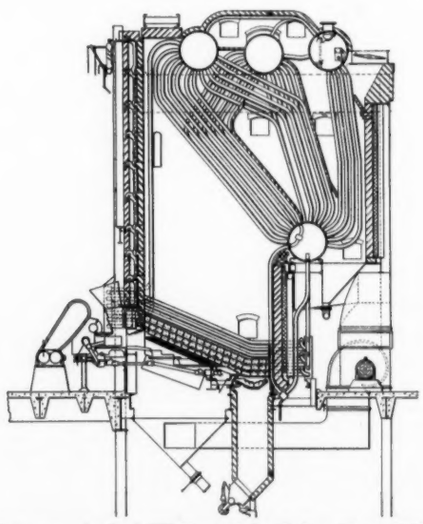

Unit comprised of CE Bent Tube Boiler (Type VA) and CE Multiple Retort Stoker. Boiler h.s.—5000 sq ft

COMBUSTION

DEVOTED TO THE ADVANCEMENT OF STEAM PLANT DESIGN AND OPERATION

VOLUME EIGHT NUMBER SIX

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EDITORIAL

Steam Generating Unit Design

Perusal of the paper by B. J. Cross in this issue will give the reader some conception of the intricate problems involved in the design of a modern steam-generating unit, with particular reference to the heat absorption. There are, of course, many other aspects such as determining circulation, proportioning various parts and figuring stresses for each particular design. Because of the many variables involved, as affected by fuel, loading and condition of the surfaces, such design, as pointed out by the author, can never become an exact science; and it is a tribute to the designers that performance usually approaches closely the predetermined figures.

Because of the rapid advance in the design of steam-generating units, data obtained through measurements taken during operation with existing units must be extrapolated into new regions. The care necessary in the collection and interpretation of such data cannot be too strongly emphasized. Despite recent advances in the theoretical analysis of heat transfer reliance must still be placed upon empirical data in the application of such analysis. This is particularly true as to furnace absorption rates.

When one considers the great variety of steam conditions prevailing and the many "tailored" designs to meet individual requirements, particularly available space in superposed installations, it is not suprising that the items of engineering time and expense frequently assume relatively larger proportions than formerly.

Domestic Oil Burners

It is not often that Annual Meeting programs of the A. S. M. E. have included papers on domestic heating, but that on domestic oil burners which was presented at the Monday evening session this year was of personal interest to many engineers because of the vast increase in domestic oil burner installations during the past few years. The paper reported tests on several widely used types, some of which showed excellent performance when properly installed in furnaces adapted to their use. The results in general, however, and the discussion of the paper form basis for the conclusion that many misfit installations exist, especially where the burners are installed in furnaces designed for burning coal, particularly anthracite. In such cases the existing combustion space is often inadequate for best results with oil and use of the wrong type of burner may give excessive flue gas temperatures. Furthermore, burners are often installed by dealers and local mechanics who have no engineering knowledge to enable them to check the individual requirements.

It is fair to assume that many engineers, fully competent in the profession that serves as their livelihood, have given little or no expert attention to their domestic fuel-burning problems. There are probably few who have applied an orsat to their flue gases, or even taken temperature measurements, and a checkup on radiation is rare. While such men exert every effort to conserve Btu for their clients and their employers they are often strangely indifferent to heat losses that affect their own pocketbooks.

Cinder and Dust Recovery

To those who attended the Panel Discussion on Cinder Catchers at the Fuels Session of the A. S. M. E. Annual Meeting it was apparent that, while much progress has been made in the development of such equipment to the end that effective designs are now available in the dry, wet and electrostatic types, nevertheless, the engineer who must make a selection is handicapped in attempting to compare performances. This is because there is, as yet, no accepted standard for determining performance. In this category is included flyash from pulverized-coal furnaces as well as cinders from stokers.

Test efficiencies that should meet the most exacting requirements were quoted, apparently in good faith, and there appeared no disposition to question the accuracy of the figures as applied to the conditions under which they were obtained; but because of the fundamental differences in the types of equipment and lack of information as to the methods of sampling employed, comparisons become difficult.

Sampling procedure is of prime importance in obtaining reliable results because large errors can be introduced by improper spotting of the samplers in the cross-section of the gas passages, especially when the dust concentration varies considerably. The means employed for separating the dust from the sample is also important. Moreover, laboratory tests may show quite different results from those obtained in actual installations.

The problem of establishing a standard method of sampling has lately been put up to a sub-committee of the Power Test Codes which undoubtedly will be aided by the vast amount of experience and data that have been accumulated by both manufacturers and operating companies, and it is anticipated that another year may see this fundamental point settled.

This, of course, is distinct from the determination of permissible stack discharge which is a problem now confronting the National Smoke Prevention Association and one in which due consideration will have to be given to coals available in different sections of the country. While the Ringlemann Chart or the Umbrascope form the basis of most smoke ordinances, they are useless in measuring the discharge of flyash.

At one time there was a disposition to soft-pedal discussions of stack discharge but this attitude is no longer general. If the public were appraised of the effort and expense that has already gone into a solution of this problem there would be a more tolerant reaction. Insofar as utility plants are concerned it should be made plain that clear stacks involve additional investment expense which properly must be reflected in rates.

HIGH-PRESSURE Installation at Mannheim

by Dipl. Ing. GERHARD KORDES
Grosskraftwerk,
Mannheim, Germany

The first superposed extension to this station was made in 1928 and consisted of three 1470-lb pressure boilers supplying steam at 895 F to two high-pressure turbines exhausting at 300 lb. Early this year a fourth boiler of 308,000 lb per hr and different design and two turbines, one back-pressure and the other condensing, were added. The station heat-flow diagram is unusual as are several features of the latest steam-generating unit. All the steam produced by this unit first passes through the fan turbine before reaching the main high-pressure turbines, and, because of the low volatile coal, the secondary air to the burners is heated to 1100 F.

INCREASING load demands in 1927 led to an extension being made during the following year to the Grosskraftwerk power plant in Mannheim, Germany, which had been placed in service four years previously. As the existing boilers operated at 300-lb pressure and 660 F steam temperature an extension employing the same pressure would not have been economical in view of advances in practice in the interim. Hence, a superposed installation was decided upon, in order to utilize the still relatively modern low-pressure turbines and their auxiliaries. Several plans were worked out and detailed calculations of their economic possibilities were studied. On the basis of these data it was decided to put in a 1470-lb installation with a steam temperature of 895 F, employing the reheat cycle. Also, in order to obtain greater economy, the superheater outlet temperature for the existing 300-lb boilers was stepped up to about 890 F.

Referring to the flow diagram, herewith, 1470-lb steam at 895 F from the high-pressure boilers[1] first goes to the two-stage steam reheater, one portion being condensed in the first stage, after raising the exhaust steam temperature from 500 to 575 F, whereas the larger portion passes directly through the second stage on its way to the high-pressure turbines. Here it gives up about 90 F in further raising the temperature of the low-pressure (300-lb exhaust) steam to 660 F. The high-pressure steam enters the high-pressure turbines at 1367 lb and 805 F and is exhausted at 300 lb. The bulk of the steam from the high-pressure turbine exhaust is reheated to 660 F and at this pressure and temperature enters the low-pressure condensing turbines. A small portion of it bypasses the reheater and flows to the high-pressure feedwater heater.

The feedwater is heated in four stages, by bled steam at pressures of 13, 66, 140 and 300 lb. Condensate from the hotwell is pumped through the heaters by a motor-driven intermediate feed pump and leaves the last heater at 300 lb and 395 F. The main boiler feed pumps, steam and electrically driven, deliver the feedwater to the economizers at about 1600 lb, makeup is provided by evaporators, and sodium phosphate is added to the feedwater ahead of the main feed pump. Sodium sulphite is also added when the presence of oxygen is suspected.

Hot-Water Storage Accumulator

A hot-water storage tank, or accumulator, of the displacement type designed by Dr. Von F. Marguerre, is placed in parallel with the last three feedwater heaters. By means of a circulating pump, more water than is required during periods of light load is heated up to 395 F and stored in the accumulator, which has a volume of 7000 cu ft. At peak loads this stored hot water augments the boiler feed and the additional exhaust steam, instead of being required for feed heating, is passed directly to the low-pressure turbines. Thus a 20 per cent increase of load is obtained for short periods without forcing the boilers.

Furthermore, a circulation storage heater is inserted in the 300-lb circuit to equalize the variations in steam pressure in the low-pressure system. Regulation of the steam pressure, feed pressure, the feedwater storage tank and firing is accomplished by an Askania control system. Also a pressure reducing valve, in parallel with the high-pressure turbines automatically functions should the latter go out of service and thus brings the high-pressure steam down to 300 lb.

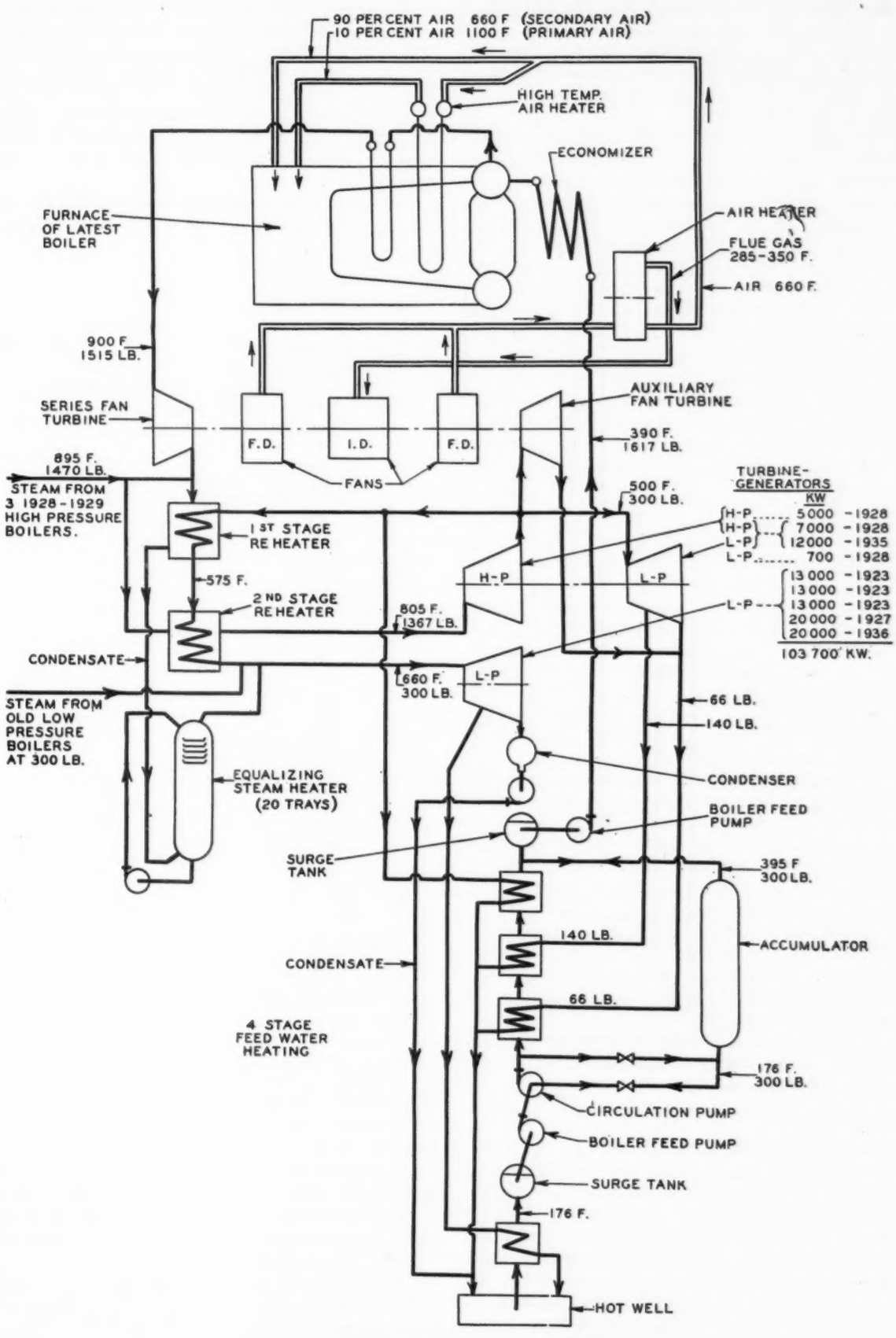
At first, during 1928, two pulverized-coal-fired boilers of a normal capacity of 140,000 lb per hr and 154,000 lb maximum were installed. One of these was equipped with an AEG system of firing and the other with a Lopulco system. A year later a third, identical boiler, was installed with Lopulco firing.

Four 200,000 lb per hr boiler feed pumps, two turbine-driven and two motor-driven, were installed to feed these boilers.

The first superposed turbine-generator was of 5000 kw capacity and the second of 7000 kw, the larger machine being arranged for regenerative feedwater heating.

After the new high-pressure extension had been in service for a short period corrosion was observed in several tube banks. This was traced to excessive oxygen in the

[1] In the case of the recently installed high-pressure boiler the steam first passes through a turbine driving the forced- and induced-draft fans.



Flow diagram of the Mannheim Station

feedwater and to poor circulation in certain tubes. By close supervision of the feedwater and the addition of sodium sulphite, the first cause was eliminated. Subsequently, circulation was improved by increasing the diameter of the downtakes and by the installation of suitable deflectors in the upper drums. Since then no corrosion has been observed.

Leakage around the handholes and at the joints of the superheated steam lines appeared after a certain period of operation. By tests and measurements it was ascertained that such leakage was caused by creep of the steel. The difficulty, in the case of the pipe joints, was overcome by substituting Van Stone flanges for the screwed flanges originally employed, and the handhole fittings were strengthened in order to reduce the stresses. Meanwhile, creep-resisting steels of chromium and molybdenum alloys became available for flanges and piping and chromium-nickel-molybdenum alloy steel for superheater handhole fittings and flange bolts. By employing these wherever possible, not only were the earlier difficulties entirely eliminated, but it also became possible to raise the superheater outlet temperature from 880 to 895 F and thus increase the overall economy of the installation.

Recent High-Pressure Boiler and Turbine

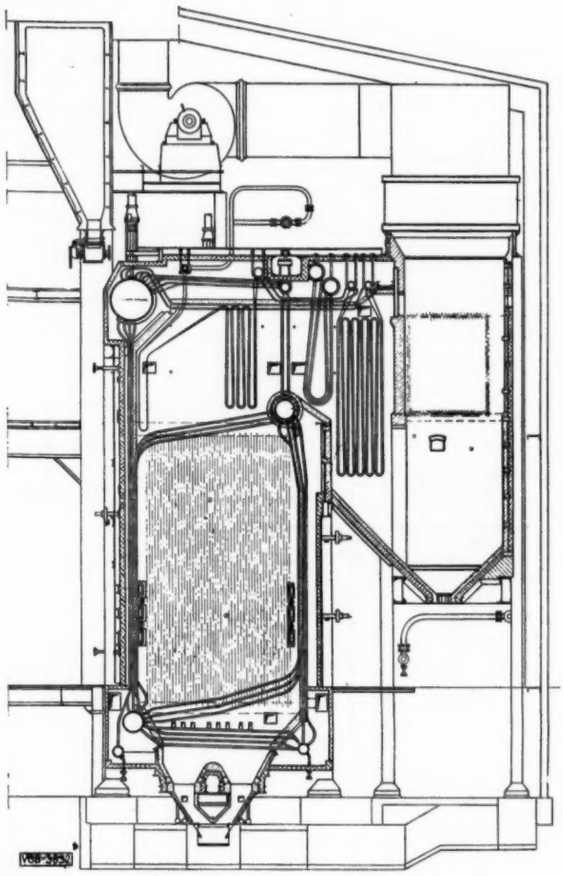
During the winter of 1935–36 a further extension was made to the station. This comprised a 308,000 lb per hr boiler, designed for 104 atmospheres (1528 lb per sq in.) and 905 F total steam temperature, to supply steam to a 12,000-kw superposed turbine-generator arranged for extraction heating; also a new 20,000-kw 300-lb condensing turbine-generator. The new boiler is pulverized-coal-fired, employing Lopulco corner firing. Because this steam generating unit has many new features it warrants a detailed description.

The coal used runs very low in volatile, 7 to 8 per cent. It is pulverized in a central mill system to a fineness 85 per cent through a 180-mesh screen. In order to burn this coal readily and economically, a high furnace temperature of 2600 to 2700 F is required. As the combustion chambers of the first three high-pressure boilers, which employed U-flame firing, were only partly cooled by water walls, the refractory walls required considerable maintenance, thus entailing both expense and undesirable outages. For this reason it was specified that the furnace of the new boiler should be completely water cooled and that the greatest possible quantity of steam should be generated in the available boiler room space. By employing corner firing and Ljungstrom air heaters the design adopted has a capacity double that of the first boilers, namely, 264,000 lb per hr normal and 308,000 lb maximum.

In order to obtain good combustion with the water-cooled furnace when burning this low-volatile coal a supplementary air heater was installed, in addition to the two Ljungstrom air heaters which preheat the total air to 660 F. This supplementary air heater, which is located between the two superheater sections, preheats 10 per cent of the combustion air to about 1100 F. It consists of tubes having hairpin bends made of chromium-silicon-molybdenum steel which is scale-resisting at temperatures up to 1300 F. To obtain quick ignition of the coal which, as mentioned, contains only 7 to 8 per cent volatile matter, an ignition zone of firebrick is provided at the elevation of the corner burners. There are twelve of these burners distributed three to each corner.

In the new boiler, which is of the K.S.G. (Cantieny) type, the dimensions of the upper drum were limited by the distance between the existing building columns which determined the width of the boiler itself and, on the other hand, by the greatest obtainable dimensions for forged drums. The upper drum has an inside diameter of 59 in. and the length of the cylindrical portion is about 26 ft.

Contrary to the customary arrangement, the circulation is such that only a part of the water circulating in



Section through latest higher pressure boiler

the boiler tubes reaches the upper drum. Water fed to the upper drum descends through the cold tubes into the lower drum, and ascends through the front wall tubes and the rear wall tubes into the intermediate drum. By suitable dimensioning of the connecting tubes, it has been possible to reduce the quantity of water reaching the upper drum to only about twice the quantity of steam generated, whereas about eight times the quantity of water, as compared to the quantity of steam, returns through the re-circulators to the lower drum and from there flows into the risers. In the upper drum and intermediate drum are inserted diaphragms in order to prevent a short-circuit between the different tube rows. The trapezoidal shape of the tube-diagram assures an equal length of the heat-exposed surface of the front wall tubes and rear wall tubes, which is also desirable.

The side wall water screens are fed by separate downtakes directly from the upper drum. Here too, re-circulating tubes are provided. The cross-sections of the connecting tubes from the upper side wall headers to the upper drum are also dimensioned sufficiently small to insure that only the double quantity of water (as compared to the quantity of steam generated) passes from the side wall screens into the upper drum. By these measures the upper drum is considerably relieved, the amount of water carried over can be easily separated by suitable deflectors and dry steam reaches the superheater.

During operation, measurements in the different tube banks have been made and it has been found that the circulation of all loads, and in particular during the starting-up period, is in perfect order and corresponds with the preliminary calculations.

As already mentioned, there are in the upper drum devices for separating the carryover from the steam. To obtain a uniform steam admission into the superheater, the connecting tubes to the superheater inlet header are uniformly distributed over the whole length of the upper drum.

The superheater itself consists of two sections, arranged counterflow with respect to the gas. In the first section, the pre-superheater, the steam is superheated to 750 F and in the second section, the temperature is raised from 750 to 895 F. The characteristic of the superheater is such that at ratings up to about 175,000 lb per hr the outlet temperature of the steam is below 895 F. In order to obtain a uniform temperature of 895 F at higher ratings up to 360,000 lb per hr, desuperheating is resorted to before the second stage of the superheater by injecting feedwater into the desuperheater. The latter is inserted in the so-called injection-header, namely, the header at the inlet end of the second stage of the superheater. Regulation of temperature is automatic by means of a thermostatic control device at the outlet side of the superheater.

To avoid overheating of the superheater during the starting-up period, a circulation pump is provided which pumps water from the lower drum into the upper drum. Thus the direction of the water flow in the superheater is directly opposite to the normal steam flow in it. The circulation pump operates until a steam pressure of about 700 lb has been reached. The further cooling of the superheater is accomplished by steam which flows in the normal direction from the upper drum to the superheater outlet, and from there over a special pipe line and pressure-reducing valve into the steam regulation accumulator. During the second period of the starting-up process excessive temperatures in the superheater are avoided by injecting feedwater into the injection header.

The heating surface before the pre-superheater is the previously mentioned high-temperature air preheater in which 10 per cent of the combustion air at 660 F from the Ljungstrom air heaters is heated to 1100 F. The ignition air flows through ducts of rectangular section arranged concentrically in the air ducts to the pulverized coal burners in the corners of the furnace.

After leaving the pre-superheater the flue gases pass over a cast-iron economizer in which the feedwater is preheated from 390 F to about 500 F. In the upper drum are arranged, before the steam separator, cascade-like plates which secure a further preheating of the water. Constant water level is obtained by means of an Askania feedwater regulator. Changes in water level, even during sudden load fluctuations of considerable magnitude do not exceed $^5/_8$ in. from normal.

Behind the economizer are installed two independently driven Ljungstrom air heaters having a diameter of approximately 16 ft in which the air is preheated from about 100 F to 660 F, whereas the gas temperatures behind the air heater vary, in accordance with the rating, from 285 to 350 F.

Unique Arrangement of Fans

The fans are arranged on top of the boiler. In the center stands the double-inlet induced-draft fan, to the right and left of which and directly coupled to it on a common shaft, are single-inlet secondary air fans. These are driven by two steam turbines. At one side is a series turbine through which flows all the high-pressure steam generated by the boiler. The pressure drop through this series turbine depends on the load on the boiler and amounts at maximum rating to about 45 lb. The nozzle sections of the turbine are such that its output is only about 80 per cent of the required power of the fans, the balance of the power being supplied by an auxiliary turbine, a so-called regulating turbine, installed on the other side of the fan group. This turbine operates with 300-lb steam and about 60-lb back pressure. By regulating this turbine it is possible to adjust the number of revolutions of the secondary air fans to the respective requirements of the boiler. The induced-draft fan has been selected for a draft which is always greater than that required at the boiler end. The adjustment of draft is attained by louvre dampers in the two flue gas ducts between the Ljungstrom air heaters and the induced-draft fan. During the starting-up period, or as long as no high-pressure steam from the boiler is available, the fans are driven by the regulating turbine, which is of such size that, in case of outage of the series turbine, it is possible to operate the boiler up to a rating of 242,000 lb per hr.

The pulverized coal is fed to the corner burners by six roller feeders. These feeders are driven by variable speed motors which receive the current from a small generator that also supplies the motors of the two Ljungstrom air heaters. This generator is driven through a shift gear by the motor of the primary air fan, which motor is on the service circuit of the power plant. In order to make the boiler independent of interruptions in the service circuit, there being no house turbine, the primary air fan is coupled with an idling auxiliary turbine which, in case of stoppage of the motor, automatically takes over the drive of the primary air fan and of the generator. As the fans, as well as the feeders and preheaters, are provided with steam drive, the operation of the boiler is not affected by disturbances in the circuit.

In conclusion, it may be mentioned that the operating results so far have checked closely with the preliminary calculations, except that at times the superheat has run too high. This has necessitated more water being injected into the desuperheater than was originally assumed. In consequence, it is intended to decrease the surface of the front section of the second stage superheater. Also, the required draft is considerably lower than originally calculated, hence the power consumption of the main fans is lower and they can be operated at speeds below 750 rpm.

Some High Spots of the A.S.M.E. ANNUAL MEETING

Brief abstracts of the papers on a new type of steam generating unit, performance of the Port Washington Station, discussion at the session on cinder dust recovery, that on embrittlement at the feedwater session and the papers on early steam turbine developments in the United States.

ATTENDANCE at the Annual Meeting of the American Society of Mechanical Engineers, November 30 to December 4, numbered 2356 which exceeded that of last year by nearly 200. The extensive program and the full attendance at most of the sessions makes it difficult to pick out the high spots of the meeting. However, from the standpoint of power engineers, particular interest appeared to center around the Monday evening paper on "The Steamotive," the paper on "The Performance of the Port Washington Station," those on "Turbine Regulation and Supervisory Control,"[1] the Panel Discussion on Cinder Catchers, the Boiler Feedwater Session and the evening session on "The History of the Steam Turbine." Several other individual papers held much interest. It was unfortunate, however, that discussion at the session on Cinder Catchers was cut short due to the room being required for another meeting.

Briefly, the Steamotive, as described in the paper by E. G. Bailey, A. R. Smith and P. S. Dickey, consists of a high-pressure boiler, auxiliaries, steam turbine and automatic control, assembled into a compact unit of minimum space and weight requirements. Although designed primarily for railway service for two steam turbine-electric locomotives of the Union Pacific Railroad, the unit also has possibilities for small industrial plant applications and marine use. The boiler, which is oil-fired, is of the forced-circulation type, of horizontal design, and has five parallel circuits arranged in such a way as to surround completely the combustion chamber. From the combustion chamber these circuits deliver the steam to a vertical separating drum in which is maintained a relatively small amount of surplus water. From this separating drum the dry steam passes to the superheater and then to the turbine. The water from the separator, designated as the "spillover," passes through a heat exchanger on its way to the hotwell. An air heater and an economizer form an integral part of the unit. Heat liberation up to 400,000 Btu per cu ft of combustion space has been obtained on test. The steam conditions are 1200-lb pressure, 950 F steam temperature at the turbine and the unit on test showed an almost straight-line efficiency curve dropping from 85 per cent at 5000-lb steam per hr to 75 per cent at 22,000 lb per hr. The fan, feed pump and fuel oil pump are all driven from a single auxiliary turbine mounted within the unit.

The Steamotive was developed jointly by the General Electric Company, the Babcock & Wilcox Company and the Bailey Meter Company, and a small unit is now in commercial service at the Lynn Works of the General Electric Company.

Port Washington Station

In his paper on the Port Washington Station of the Milwaukee Electric Light & Railway Company, F. L. Dornbrook briefly described this 1390-lb, 80,000-kw, single-unit station, and gave operating data and experience covering the period from October 1935, when the plant was placed in service, through August 1936. During this period the average net heat consumption has been 11,220 Btu per kwhr, with a low month in July of 10,857 Btu; that for the past three months being well under 11,000 Btu. This station operates on the reheat cycle and superheat is provided by a convection and a radiant superheater in series. It was of particular interest that the charts reproduced showed an almost constant superheat over load variations of 3 to 1.

Such outages as have occurred to date have been due chiefly to shutting down for scheduled inspections, for lack of load, for a change in the amount of reheater surface during the second month of operation, for adjusting the turbine governor and for the replacement of gaskets. The availability of the plant as a whole has been 92.6 per cent.

Because of the large furnace and the relatively low heat release of about 17,000 Btu per cu ft, no slagging troubles have been encountered and the first row of boiler tubes has never been lanced.

In the discussion of Mr. Dornbrook's paper, H. G. Thielscher of the Potomac Electric Company stated that the Buzzard Point Station, which has two 650-lb pulverized-coal-fired boilers of the slagging-furnace type supplying steam at 825 F to a single turbine, had averaged a heat rate of 12,550 Btu for the year 1935, with a corresponding load factor of 62.8. Because of the erosion of the metal blocks covering the furnace walls, these are now being replaced by stud tubes. The longest single run has been six months, three-and-a-half of which was at full load, and at times the load range has been as great as 7 to 1. Low load operation has not resulted in erosion of the air heaters because the preheated air is partly recirculated.

Cinder Catcher Session

At the Panel Discussion on Cinder Catchers, brief papers were presented by W. G. Christy, who dealt with the problem from the smoke inspector's standpoint; H. L. Whiton, who described and gave performance figures for the Thermix Recirculator Dust-Collector;

[1] See abstract elsewhere in this issue.

Stanley Brown, who discussed the Van Tongeren cyclone-type of dust catcher employing the eddy current principle; Allison Craig, who dealt with the Riley wet-plate washer; G. C. Derry of the B. F. Sturtevant Company, who discussed the cinder-vane type of fan; M. D. Engle of the Edison Electric Illuminating Company of Boston, who told of his company's experience with the Pease-Anthony wet-type scrubber at the Kneeland Street Heating Plant in Boston; J. J. Grobb of the New York Edison Company, who related experiences with the wet-type cinder-catcher at the Hell Gate Station; H. P. Hardie of the Brooklyn Edison Company, who discussed sampling flue dust; J. R. James of the Detroit Edison Company, who told of the utilization of fly ash in making building block and a new type of concrete; and also a representative of the Research Corporation, who spoke on the electrostatic arrester, particularly the new type in which a half wave length is employed for precipitation.

The performances reported with all types were excellent and it was unfortunate that time did not permit further elucidation through questions from the audience. It was emphasized by Mr. Hardie, however, that much depends on the method of sampling and in the absence of an accepted standard it is impossible to compare performances. It seemed to be the consensus of opinion that wherever possible the cinder-catcher should be placed after the air heater in order to take advantage of a more uniform distribution of the flue gas, and before the fan in order to cut down erosion of the blades.

In the commercial use of fly ash, as related by Mr. James, the concrete is made up of a mixture of 100 lb of fly ash, 100 lb of portland cement and 250 lb of cinders. This, at the end of 28 days, forms a very satisfactory concrete which has a strength of 3000 lb per sq in. and an absorption of less than 6 per cent. The cost of this concrete is about the same as that of ordinary gravel or stone concrete. In fact, the Detroit Edison Company is about to construct a six-story office building in which it will be used for the floors.

The speaker also described the manufacture of Rostone block which is made from fly ash, rosin and lime, molded in a special machine having small clearances. This block requires fifteen hours to dry in a steam kiln and is suitable for both interior and exterior use. It has good weathering qualities, can be carved and sawed, and possesses a high fire resistance.

Boiler Feedwater Session

It will be recalled that last year at this session the rather startling announcement was made of evidence to indicate that the presence of sodium silicate in sodium hydroxide solutions was responsible for embrittlement. Subsequent investigations carried on at the New Brunswick Station of the Bureau of Mines have been directed toward ascertaining the action of sodium silicate in this connection. However, inasmuch as the work is still in progress, and definite conclusions have not yet been arrived at, the report is not published, but W. C. Schroeder of the Bureau of Mines reported briefly on progress to date.

These experiments have been conducted with iron powder immersed in the sodium hydroxide and sodium silicate solutions. It would appear that sodium hydroxide solution acting on iron powder at around 480 F for 16 hours releases a considerable quantity of hydrogen and that this increases with the concentration of the solution. When small quantities of sodium silicate are added there is a sharp rise in the evolution of hydrogen during the first few hours, after which there is a sharp decline. Further increase in the silicate concentration does not appear to affect the liberation of hydrogen. The concentration of sodium silicate at which a marked decrease in the production of hydrogen is noted appears to correspond to that at which embrittlement starts.

Early Turbine Developments

The group of papers dealing with early turbine history in the United States provided a fascinating account of pioneer developments, failures as well as successes, patent situations, the transition stage from engines to turbines and the rôles played by many well-known personalities. Commercial development of steam turbines abroad antedates that in the United States, hence most of the earlier designs in this country were predicated on foreign turbines.

Entrance of the Westinghouse Company into the turbine field dates back to 1895 when George Westinghouse became interested and took out a license from Sir Charles Parsons in England, who at that time had built some 400 machines in sizes up to 400 kw. The license agreement provided that a turbine generator set be shipped from England to Pittsburgh as an example of construction. This was a 150-kw unit of the reaction type, and after it had served this primary purpose it was presented to Cornell University, but has lately been removed to the Museum of Science and Industry in New York City. Coincident with taking out the license, Francis Hodgkinson was released from the Parsons Company and became associated with Mr. Westinghouse.

The first turbine constructed by the Westinghouse Company was produced in the Spring of 1896 for the Nichols Chemical Company in Long Island City. This was a 120-kw, 180-volt d-c generating unit designed for 500 rpm, but because of commutator troubles at this speed it was rejected by the customer and returned to East Pittsburgh where changes were made, and it was subsequently used as an exciter. It was later broken up.

A somewhat epoch-making turbine, because of its large size at that time, was built for the Hartford Electric Light Company in 1900. This was a condensing unit of 1500-kw capacity, of the single-cylinder type and operated at a speed of 1200 rpm and 155-lb steam pressure. It was scrapped in 1908 as it was found to suffer from lack of reliability.

The first tandem-compound unit was built by Westinghouse in 1901 and sixteen of this type were installed up to 1904 ranging in capacities from 750 to 1500 kw.

Around 1907 design changes were initiated in order to make possible capacities up to 10,000 kw at standard synchronous speeds and an impulse stage was substituted for the small diameter reaction elements in the first section; also, the double-flow design was applied to the low-pressure section. A few years later blades and blade fastenings were completely re-designed, and the extraction type of turbine was first turned out in 1911.

Professor A. G. Christie of Johns Hopkins University, who from 1905 to 1907 was associated with the turbine department of the Allis-Chalmers Company, told of the early developments of that company in the turbine field. This dates back to 1902, when the company engaged

Charles Backstrom to design a turbine of the radial-flow type bearing his name. This machine, however, turned out to be a failure and negotiations were next entered into with Escher-Wyss Company of Zurich, Switzerland, with a view to securing a license to build the Zoelly turbine. These negotiations fell through, however, as the type under consideration proved unsatisfactory and Zoelly changed to the Rateau design. Following this, various other foreign designs were investigated and subsequently several six-stage turbines were built from the designs of Max Patitz, consulting engineer of the Allis-Chalmers Company. However, it subsequently became possible to secure license under the Parsons patents, and the Patitz design was abandoned.

The first turbines built by the Allis-Chalmers Company were ordered through the late Thomas E. Murray in 1904. These were 1500-kw machines and were installed at the Washington Street Station of the Utica Gas & Electric Company, the North River Station of the New York Edison Company, the plant of the Westchester Lighting Company and in the station of the Dayton Power & Light Company. A larger machine of 5000 kw was installed in 1905 at the Kent Avenue Station of the Brooklyn Heights Railroad Company which had recently been built under the direction of Mr. Murray and George A. Orrok, then mechanical engineer of the New York Edison Company, who was assistant to Mr. Murray.

The third paper of this group, by E. L. Robinson, told of the early turbine developments by the General Electric Company. This dates back to 1897 when that company entered into an agreement with the inventor of the Curtis steam turbine, the basic patent of which provided for pressure staging, using the impulse principle with one or more rows of moving buckets. Following this a number of experimental machines were built and tested at the Schenectady Works.

In 1901, W. L. R. Emmet and Oscar Junggren were associated in the design of the first 500-kw turbine which was of the horizontal type and had two multiple-row wheels in separate casings. This machine, which ran at 1200 rpm, began regular service at Schenectady in 1902. Shortly thereafter a similar machine, rated at 1500 kw at 720 rpm, was built for the Port Huron Light and Power Company which later became a part of the Detroit Edison system.

The manufacture of vertical turbines in capacities of 500, 1500 and 5000 kw was begun in 1902, and the first machine of this type was placed in commercial service in 1903 at the plant of the Newport and Fall River Company, later the Bay State Street Railway Company. It was of 500-kw capacity at 1800 rpm. This was followed closely by a 5000-kw, 500-rpm machine installed at the Fiske Street Station in Chicago.

The author next reviewed the large number of turbine inventions which Oscar Junggren patented from 1903 to 1907 including the principle of the automatic governor which still characterizes all large G.E. turbines.

By 1905 there were in operation 224 turbines of the larger size, aggregating 350,000 hp, including ten Curtis turbines of 5000 kw.

While the Curtis patents covered single-row wheels as well as multiple-row wheels, the success of the early machines was attributed to their incorporation of the multiple-row feature.

In 1908, a 1500-kw, 1800-rpm four-stage horizontal turbine-generator was completed for the Cranford, N. J. Station of the Public Service Electric Company. The first 20,000-kw vertical-shaft machines were built in 1911 and a number were installed in utility plants during the next two years, but in 1913 this type was discontinued. There was then developed in their place a fairly definite design for horizontal turbines having seven to twelve stages, the first having two rows of buckets with nozzle governing, and the others single-row wheels with little change in pitch diameter. The first 20,000-kw horizontal machine was shipped to Chicago in 1913 and later in the same year the New York Edison Company ordered a 30,000-kw machine.

It was pointed out that the development of General Electric turbines has been essentially along American lines independent of foreign designs, the single exception being a short-lived effort to exploit the Ljungström turbine in this country. This effort was abandoned following the bursting of one of the exhaust wheels of a 15000-kw machine while on test at Schenectady. Moreover, it had been found that the Ljungström turbine was expensive to build under American factory conditions and it was not clear that it was more efficient than multistage impulse machines.

Combustion of Solid Fuels

It is now possible to calculate approximately what happens to a piece of fuel as it undergoes consumption in a fuel bed, without actually building a furnace and burning the fuel and measuring what happens. This, according to M. A. Mayers of the Carnegie Institute Coal Research Laboratory, means that fuel engineers can predict quantitatively the requirements for good combustion—the factors which have been known for many years as "the three T's," namely, time, temperature and turbulence. In other words, it implies a high enough temperature so that burning can proceed rapidly, in a turbulent, or well-mixed atmosphere, for a long enough time so that the fuel would be burned completely to ash.

These requirements are based largely on the earlier tests of Hood and Kreisinger at the Bureau of Mines, before and during the war, and result from their observations that a considerable furnace volume was required for complete combustion. Until very recently the results of these had not been analyzed quantitatively and the engineer's experience was the only guide to a quantitative estimate of *what kind of turbulence*, at *how high a temperature*, for *how long a time* were required for complete combustion.

A first approximation to an estimate of the quantitative relations among "the three T's" is now possible. The speed of the chemical reactions, and its dependence on the temperature have been investigated in the laboratory. This determines the temperature requirement. It is also found that the state of motion of the air flowing through the fuel bed, whether it be fast or slow, or whether it flows smoothly or in eddies and whirls, helps to determine the speed of burning. Thus, study of this factor makes it possible to predict the turbulence requirement. Finally, when these two factors are known the speed of combustion can be calculated and thus the time required for complete combustion can be determined.

Problems in Collection and Evaluation of Data for the Design of Steam Generating Units*

By B. J. CROSS

Combustion Engineering Company, Inc.

The author reviews the various factors involved in the design of a steam generating unit, showing that because of variations in the fuel burned and in the condition of the heating surfaces, such design can never be an exact science but must be dependent to a large extent on experience. Formulae are given for the transfer and absorption of heat, and the measurement of temperatures by means of thermocouples is discussed. Curves are included to illustrate heat absorption and temperature gradients for given types of furnaces and methods of firing.

THE performance of a steam generating unit is "built" into it by the design, and is fixed by the arrangement of heating surfaces and the path of gases over these surfaces. If the surfaces could be maintained clean and the gas flow distributed uniformly the performance could be accurately predicted. Uncertainties of performance are the result of fouling of the surfaces and unequal distribution of gases.

Rational progress in the development of steam generating units must be made step by step, the operating experience of one design leading to the development of the next. It is obvious, however, that if there is to be progress, ventures must be made into new territory. The hazards of failure of such ventures can be greatly reduced by careful deductions from known facts and extrapolation of actual operating data. The purpose of this paper is to indicate some of the difficulties encountered and the care necessary in making field measurements in obtaining such operating data. For the determination of overall performance of a steam generating unit only end-point data are necessary; that is, heat input, heat output and heat losses only need be considered. For the purpose of obtaining design data, measurements throughout the unit are required and the performance of each part of the unit must be carefully studied.

The rate of heat transfer from a hot fluid to cooler absorbing surfaces may be calculated with a satisfactory degree of accuracy when the mass and the temperature of the fluid are known, when the flow of fluid is uniformly distributed over the heating surfaces and when these surfaces are clean. These ideal conditions never occur in practice, and the designer must consider both theoretical and practical data. In general, the shape or slope of an absorption curve may be determined from a theoretical consideration but empirical constants must be established to locate the curve with respect to its reference axes.

The problem of design of a steam generating unit begins with the furnace. It is there that the heat latent in the fuel must be developed for delivery to the boiler. In the not far distant past, the sole function of the furnace was considered to be the evolution of heat and any absorption of heat that occurred was merely incidental to the exposure of boiler tubes to the furnace. Water-cooled surface was first introduced into the furnace to protect refractories from erosion by molten ash. It next replaced refractory surface and in the high capacity furnace of today, refractory surfaces are almost completely eliminated and the absorption of heat has become an important function. In units of recent design 40 to 50 per cent of the available heat in the fuel is absorbed in the furnace.

From a consideration of combustion only, the rating or duty of a furnace is expressed by the heat liberated per cubic foot of combustion space and the effectiveness of the furnace is measured by the composition of gases delivered to the boiler. Considered as a heat absorbing unit the rating of a furnace is the heat liberated per square foot of radiant heat absorbing surface and the measure of its performance is the temperature as well as the composition of the gases leaving the furnace.

The area of radiant heat absorbing surface may be calculated in various ways. It may be based on the entire circumference of furnace tubes, on one-half the circumference, on projected surface of the tubes only or on the projected surface of tubes and appurtenances such as fins, pegs or attached blocks. As long as absorption rates are determined on the same basis, it does not matter how the radiant heating surface is computed. Projected areas are recommended for simplicity of calculation. A large part of the discrepancy in published values for furnace absorption rates is due to differences in methods of computing radiant heating surface. Convection heat absorbing surface is fixed by official definition. Radiant heat absorbing surface should also be officially defined.

* A paper presented at the Annual Meeting of the A. S. M. E., New York, November 30 to December 4, 1936.

The transfer of heat in the furnace is principally by radiation and takes place according to the Stephan-Boltzman law involving the difference in the fourth power of the absolute temperatures of the radiating surfaces. As applied to heat absorption in a furnace, the equation of this law may be written:

$$Q = C_1A_1T_1^4 - C_2A_2T_2^4$$

where Q is the quantity of heat absorbed, C the radiation constant including the emissivity factor, A the surface area and T the absolute surface temperature. The subscripts $_1$ and $_2$ refer, respectively, to the hotter and the cooler surfaces. This expression treats each surface separately as radiating surface and assumes the difference in heat radiated to be the net heat transferred.

The factor C_1 can be based on an emissivity factor of unity without serious error. The area A_1 will vary with the method of firing and also with the rate of firing and percentage of excess air. If we consider the outer envelop of the flame as the active radiating surface, the area A_1 will always be smaller than A_2 which is its limiting value. The temperature T_1 does not exist as such. It is the average temperature corresponding to the average rate of absorption. The factor C_2 for the wall surface depends upon the type of wall construction and upon its state of cleanliness. The area A_2 is fixed by the design. The temperature T_2 of the wall surface is controlled in some measure by the design pressure but is greatly affected by the state of cleanliness. For a clean tube surface the temperature T_2 cannot greatly exceed the temperature corresponding to the operating steam pressure but if covered with ash it can assume any temperature up to the fluid temperature of the slag. When boilers are operated at constant high ratings, wall tubes may become encased in a thin layer of almost vitreous slag which must be close to the molten temperature on the outer surfaces. When the rating is lowered, these accumulations fall off. When the boiler is operated with variable rating, slag is constantly building up and falling off so that furnace surfaces remain comparatively clean.

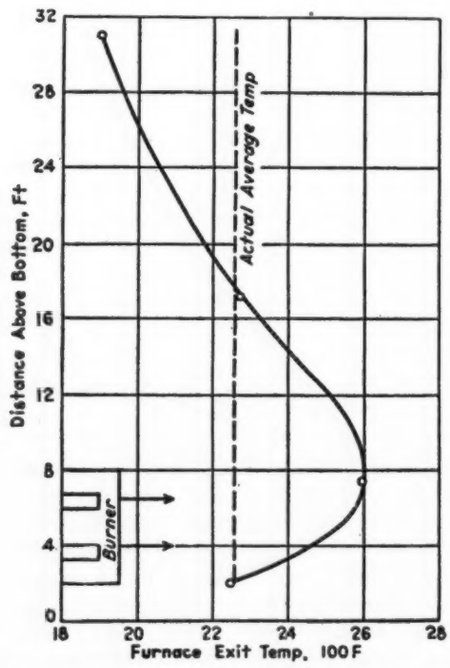


Fig. 1—Furnace-temperature gradient for a completely water-cooled furnace fired with powdered coal

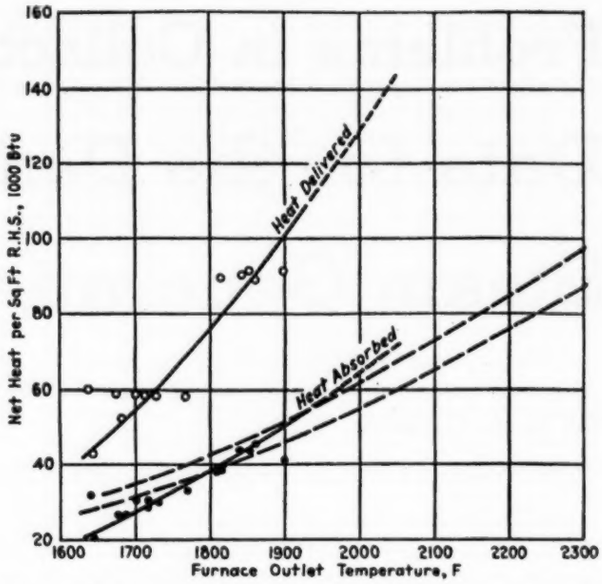


Fig. 2—Firing and absorption rates in a water-cooled furnace fired with powdered coal

In Fig. 1 a furnace temperature gradient is shown for a completely water-cooled furnace fired with pulverized coal. The burners are arranged tangentially at the bottom of the furnace. The temperatures shown were measured with platinum-platinum-rhodium thermocouples inserted through openings at different levels in the furnace. They are average readings over an insertion distance of about 8 ft. At the lower part of the furnace they may be correct within 100 deg. The theoretical flame temperature was 3600 F while the maximum measured temperature was 2600 F. The theoretical temperature of flame cannot be attained unless the combustion of the fuel is instantaneous. Actually, a definite time is required for combustion and radiation occurs during combustion so that the theoretical temperature is never reached.

The only furnace temperature which can be measured with any degree of accuracy is that of leaving gases and most of the methods that have been proposed for computing furnace absorption rates are based on the Stephan-Boltzman law using the furnace outlet temperatures.

In Figs. 2 and 3, firing rates and absorption rates are plotted against furnace outlet temperatures. Fig. 2 shows curves for a completely water-cooled furnace, tangentially fired with pulverized coal. This is the unit, the furnace temperature of which is shown in Fig. 1. Fig. 3 refers to a vertically fired unit also completely water cooled. For the purpose of comparison, the Stephan-Boltzman curve based on furnace outlet temperatures is shown in each figure. The heat rates shown on the ordinates represent the net heat per square foot of radiant heating surface. This net heat is equal to the gross heating value of the fuel, plus the heat in the preheated air, less the latent heat of water vapor and the heat in unburned carbon. It will be noted in both of these figures that the actual absorption rate has a steeper slope than the Stephan-Boltzman curve. This may be accounted for by the increasing value of A_1 with higher

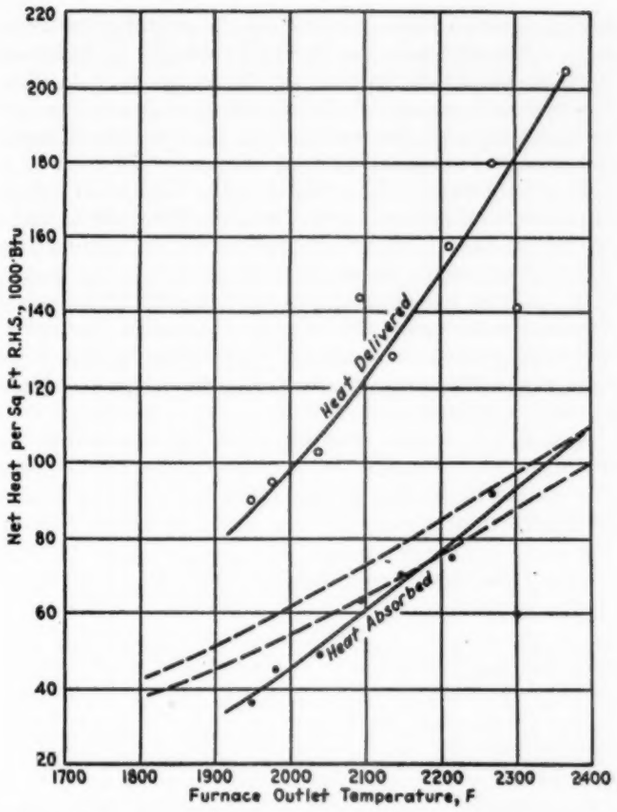


Fig. 3—Firing and absorption rates in a water-cooled vertically fired furnace

combustion rates and also by an increase in the convection component with the increased velocity of gases in the furnace. These figures indicate the difficulties in applying the Stephan-Boltzman law to the solution of problems in furnace heat transmission. That these difficulties may not be unsurmountable is indicated by some of the recent treatises on the subject.

Measurement of the temperature of a stream of gases adjacent to or surrounded by cooler surfaces is liable to a radiation error. Theoretically, with any instrument having a mass exposed to the gases, the true temperature will never be indicated. The instrument is receiving heat by conduction and radiation from the gases and is losing heat by radiation to the cooler surfaces. It will finally assume an equilibrium temperature somewhat different from that of the gases. The smaller the mass of the instrument, the closer will be the equilibrium temperature to the true gas temperature. Because of the small mass possible, the thermocouple is well adapted to the measurement of gas temperatures. For the furnace outlet temperature platinum-platinum alloy couples are necessary, preferably in water-cooled mountings. With such couples wire diameters as low as 0.005 in. may be used. The use of small wire reduces the radiation error and also the error due to conduction of heat along the wire to the water-cooled mounting.

One difficulty experienced with thermocouples is that due to slag building up on the wire and increasing the effective diameter, thus causing a false reading because of the shielding effect and increased radiation. The couple must be frequently removed and cleaned. Couples of the "velocity" type are sometimes used. Such couples have a refractory shield surrounding the hot junction to reduce the radiation and are also arranged so that gas may be drawn over the junction at relatively high velocity. Such couples give good temperature readings while they remain clean but they foul rapidly and require frequent cleaning. This is particularly true in pulverized coal furnaces.

As a thermocouple located at the furnace outlet is exposed to radiation from the hotter furnace below it, as well as to the cooler surfaces above, it is questionable whether the radiation error will be plus or minus. In several cases where the temperature was measured with two couples, 0.025 and 0.012 in. diameter, the larger couple gave the higher reading though the differences were not great. After experience with both shielded and bare couples, the preference of the writer is for the bare couple. It is much easier to clean and does not require cleaning as often as the velocity couple.

As the gas temperature measurements at the furnace outlet are at the largest flow section of the unit, a large number of points must be taken to get an average and considerable time is required for a traverse. While, fortunately, most large steam generating units can be operated under quite steady conditions for periods long enough to make complete measurements over the entire unit, care should be taken that a change of operating conditions does not occur while the traverses are being made.

Ordinarily, the arithmetical average of the individual points on a temperature traverse of the furnace outlet may be considered to be a true average temperature. If there are wide differences in individual readings, differences in gas velocity may be suspected. If, however, the temperature is reasonably constant over the section, differences in gas velocity need not be considered.

The heat transfer for all surfaces above the furnace is chiefly by convection, radiation transfer becoming less and less a factor as the temperature of the gas is reduced. The factors controlling the convection transfer rate have

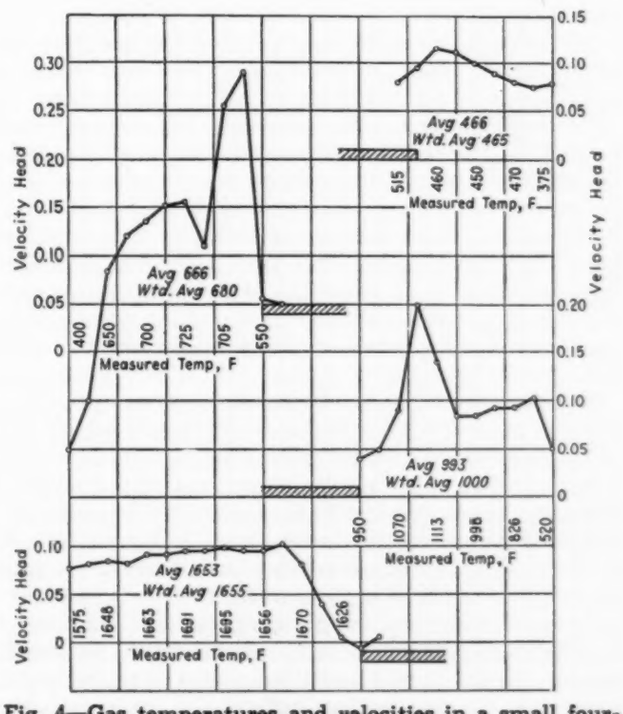


Fig. 4—Gas temperatures and velocities in a small four-pass marine boiler

been the subject of much study and theoretical values can be calculated that have a good agreement with practical values. Heat transfer by convection is represented by the equation:

$$Q = RS\,(tm)$$

R being the convection transfer rate in Btu per square foot per hour per degree F, S the surface in square feet and tm the mean temperature difference between the gases and the surface. The heat transferred by convection also must equal the product of the weight, the specific heat and the temperature drop of the gases or

$$Q = Wcp(t_1 - t_2) \text{ and } R = \frac{Wcp(t_1 - t_2)}{S(tm)}$$

For the experimental determination of R, the temperature and weight of gas are required. An inspection of the last equation discloses that the error in the determination of W has a direct effect on the value of R. As a difference in temperature appears in the numerator consistent errors in temperature measurement are largely compensating although in the denominator they affect the value of R inversely with the sign of error. As consistent errors in temperature measurement are more likely to be minus than plus because of radiation error and also because of variation in gas velocity, the determined values for R are likely to be high.

Unlike measurements at the furnace outlet, the radiation error in gas temperature measurements in the boiler passes must always be negative as the instrument used is completely surrounded by surfaces at a lower temperature than that of the gases. The radiation error is less at high boiler ratings when the gas velocity is high. Shielded "velocity" thermocouples or thermocouples made with small wire and small hot junctions should be used. The small wire bare thermocouple is recommended for its simplicity. At the ratings at which present-day units are usually operated, the radiation error should not be large.

The effect of varying velocity when the arithmetical average of a number of readings is taken is to cause a minus error in the temperature. The transfer rate R varies with a power of the mass flow. If this exponent were unity, the transfer rate would increase at the same rate as the flow and the temperature of gas leaving a boiler would remain constant regardless of rating. Actually, the exponent is somewhat less than unity and the gas temperature increases with flow. The points of high velocity therefore become points of high temperature and if given equal weight as in an arithmetical average, the average gas temperature would be in error on the low side.

Bulletin 214 of the U. S. Bureau of Mines describes tests on a small marine boiler during which special investigations of gas temperature and gas velocity were made. In Fig. 4 the results of one test on a four-pass boiler are shown graphically for each of the four passes. The arithmetical mean temperature is compared with a weighted mean based on velocity and density. It will be seen that in the first pass, where there is no relation between velocity and temperature, there is practically no difference between the arithmetical and the weighted means. In the second and third passes after the gases have passed over heating surface, the high velocity points become high temperature points and the weighted average gives a higher value than the arithmetical average. The difference in the two averages is, however, surprisingly small considering the range in velocities. In the fourth pass there is less variation in velocity and temperature and the arithmetical and weighted mean temperatures are practically the same.

In most instances the arithmetical average of a number of readings taken over a section of gas flow will be sufficiently accurate. When a high degree of accuracy is required or when great differences in velocity occur, velocity head and temperature readings should be taken simultaneously and the weighted average obtained. The design of a combination thermocouple and pitot tube is not difficult particularly as only relative and not absolute velocities are required.

The weight of gas resulting from the combustion of fuel may be determined by means of a carbon balance. The weight of dry gas per pound of fuel is given by the familiar formula

$$W\,(\text{dry}) = \frac{4CO_2 + O_2 + 700}{3\,(CO_2 + CO)} \cdot C_b$$

where the various gases are in per cent by volume and C_b represents the pounds of carbon burned per pound of fuel. To obtain the total weight of gas per pound fuel there must be added to the dry gas the moisture in the fuel, the water vapor formed from burning hydrogen and the water vapor in the air used for combustion.

To obtain a representative value for the composition of gas at a flow section of the boiler, samples must be taken at a number of points corresponding to the points for temperature measurement.

Four Gas Sample Traverses Necessary

Except for the fact that air infiltration occurs throughout the unit, only one determination of gas flow would need to be made for studies of the entire unit. Air infiltration increases with draft and is, therefore, likely to to be higher in the latter stages of heat absorption, that is, in the economizer and air heater. For a complete study of the steam generating unit, four gas sample traverses are necessary, one at the furnace outlet in connection with studies of the furnace, superheater and boiler surface below the superheater, one at the boiler outlet, one at the economizer outlet and one at the air heater outlet. For each of these sections of heating surface beyond the first pass, the average of the inlet and outlet gas weights should be used. As in making temperature averages, differences in gas velocity must be considered in making up the average of the analysis of gas samples and if great differences in gas velocity are found to exist, weighted averages should be used.

The rate of heat transfer in that portion of boiler surface below the superheater is of interest chiefly in the superheater design. The temperature of the gases leaving the furnace must be reduced to accord with the degree of superheat required and also to comply with the limit set by the temperature tolerance of the metal of the superheater elements. With the present-day demand for higher superheat temperature and with the development and use of high temperature alloys, the amount of surface below the superheater is becoming smaller and wider tube spacing is being used. This smaller surface with wider tube spacings presents new factors in the mathematical determination of heat trans-

fer rates which are perhaps not covered in existing formula and new experimental data are required.

The superheater may be either of the convection or radiant type or a combination of both types. Radiant superheaters are located in the furnace and may be treated in a way similar to water-cooled surface. Convection superheaters are located in the path of gases after a part or all of the boiler surface. The calculation of the heat transfer rate for a superheater is more involved than for water-evaporating surfaces largely because the mean temperature difference is difficult to determine. The multiloop design of large superheaters does not permit either a true counterflow or a true parallel-flow arrangement. For this reason, superheater transfer formulae are almost entirely based on empirical data.

Economizers, while designed on a counterflow principle, are not, strictly speaking, true counterflow, but similar to the superheater are partially crossflow. However, because of the comparatively high heat capacity of water, there is only a small water temperature rise over the length of a single tube and for practical purposes, economizers of the usual design may be considered truly counterflow.

Gas temperatures and gas weights for the economizer may be determined as heretofore described without much difficulty. The sections of gas flow are smaller than for boiler passes and the traverses are correspondingly simplified. The heat absorbed by an economizer may be determined both by the gas temperature drop and by the water temperature rise, one method serving as a check upon the other. The latter method probably is the more accurate as water temperatures may be measured more accurately than gas temperatures.

Heat Transfer in Economizers

The transfer of heat in the economizer more than for any other part of a steam generating unit, lends itself to mathematical analysis. When the economizer elements have extended surfaces, the effectiveness of such additional surfaces must be determined experimentally. A factor may then be applied to the total surface to convert it to effective tube surface.

Air heaters are of three general types, the plate or envelope type, the tubular type and the regenerative type. The plate and tubular heaters are similar in principle differing chiefly by the method of confining and separating the air and gas. In the regenerative air heater, the heat of the gas is first imparted to a solid. This solid is then transferred mechanically to the air stream and the heat is given up to the air. The principle of heat transfer is not greatly different from that in the other types of heaters but the temperature gradients are somewhat involved and difficult to define. An exact mathematical analysis and derivation of air heater transfer rates is impractical and design formulae are based largely on empirical data.

Air heaters are always designed on the counterflow principle but because of entrance and exit requirements, a part of the path is crossflow. The longer the heater can be made, the more nearly it approaches true counterflow. In all air heaters the transfer of heat involves two gaseous mediums and the overall transfer rate from the gas to the air is a combination rate based on the transfer rate of the gas to a solid and from the solid to air. If R_g is the gas transfer rate and R_a the air transfer rate, the net rate of transfer R from the gas to the air is expressed by

$$\frac{1}{R} = \frac{1}{R_g} + \frac{1}{R_a}$$

While this principle holds true for all transfer rates throughout the boiler, the transfer rate on the water sides of tubes is so high and the value of its reciprocal so low that it may be neglected.

The heat absorbed by an air heater is determined by the weight and temperature drop of gas passed through it. If the weight of air passing through the heater could be determined accurately, it would serve as an independent check on the heat absorbed. Because of air infiltration ahead of the air heater, the weight of air passing through the heater is always less than the amount indicated by the analysis of gases passing through the heater.

Effect of Velocity Through Air Heater

The temperature of gases to and from the heater and the temperature of air from the heater may be determined by temperature traverses at sections of flow. Because the encasing walls of ducts and flues absorb but little heat, their temperature is not greatly below that of the gases and there will be no appreciable error due to radiation. Differences in velocity, however, must be considered. High gas velocity points become high temperature points, as has been previously described. High or low gas temperature points may, however, occur independently of gas velocity because of differences in air velocity. On the air side of the heater, high velocity points become low temperature points and high and low temperature points may also occur independently of air velocity, being caused by differences in gas velocity. Ordinarily, air heaters are designed in such a way that large differences in velocity of either gas or air cannot occur, but sometimes the design is affected by mechanical interferences and space limitations and unfavorable conditions for gas and air flow may result.

The design of steam generating units will probably never approach anything like an exact science. The behavior of water in its liquid and vapor forms has been thoroughly investigated and accurate tables of its properties have been prepared. Because of its high heat capacity and high density it is ideal as a carrier of energy. It can be confined to small tubes and its flow directed, if not with accuracy, at least with certainty. As an absorber of heat it leaves little to be desired.

On the other hand, fuels are many and varied. They may be solid, liquid or gaseous. They vary widely in physical properties and chemical composition and contain many impurities. They must be burned with air which contains 80 per cent inert substance. The hot gases derived from their combustion have low heat capacity and low mass. Because of the energy cost in moving so tenuous a medium, velocities are limited and large flow sections must be provided. These gases must be passed in series over each of the several parts of the steam generating unit. The performance of each part thus is affected by the performance of all preceding parts.

Considering the difficulties involved, the actual performance of steam generating units in general compares favorably with design performance. That occasional departures occur indicates that all factors are not completely under control. Emphasis should be made, not on occasional misses, but on the large number of hits.

NEW CATALOGS AND BULLETINS

Any of the publications will be sent on request.

Boiler Water Treatment

Boiler water treatment has become such a complicated subject that a new 16-page bulletin entitled "The Inside Story of Boiler Water Conditioning," issued by the Elgin Softener Corporation, Elgin, Ill., will be welcomed by industrial executives, engineers and power plant operators. This bulletin discusses boiler water treatment in a clear, non-technical manner, shows the importance of selecting the right method to handle the conditioning of various waters and emphasizes the necessity for a thorough understanding of the subject if the boiler room is to be efficiently operated.

Building Maintenance

Of interest to plant engineers is a 40-page booklet issued by the Flexrock Company, Philadelphia, dealing with various phases of building maintenance, including water-proofing, floor coverings, roofing, etc. The booklet is fully illustrated and contains much useful information.

Coupled Pumps

Bulletin No. 7066 has just been issued by the Cameron Pump Division of Ingersoll-Rand Company, New York, and covers its line of coupled pumps in capacities ranging from 150 to 5000 gpm operating against heads between 20 and 250 ft. These pumps are employed for handling various liquids, in addition to water. The bulletin contains numerous views of installations as well as construction details of the coupled pump.

Electric Machinery Catechism

Fairbanks, Morse & Co. has just issued a 48-page booklet entitled "Electric Machinery Catechism" which, in question and answer form, deals with the fundamentals of electric circuits, the principles of direct- and alternating-current generators and motors, describes the various types of motors, their characteristics and applications, and finally discusses electrical machinery in general. Simple diagrams are used freely as part of the explanations. Several useful tables are included. The text is essentially non-technical and should be very useful to the mechanical engineer whose knowledge of electricity is limited or who has become rusty in such knowledge.

Flow-Meter Handbook

The 1936 edition of this handbook, compiled by Louis Gess and J. O. Feldmark of the Flow-Meter Engineering Department of The Brown Instrument Company, contains 162 pages covering such subjects as flow-meter engineering, orifice design, steam-, water-, liquid-, air- and gas-flow measurements, general tables and engineering data. The text is fully illustrated, numerous useful charts are included and examples are worked out. The book is substantially bound and is priced at $2.

Furnace Constructions

Bulletin No. 101 discusses and illustrates standardized furnace walls as constructed by the George P. Reintjes Company, Kansas City, Mo. These walls are sectionally supported and the refractories are fused into solid units between the expansion joints by use of air-setting high-temperature cement, thus eliminating air leakage. They may be placed behind water walls and the tiles are supported on box castings.

Hydroseal Pumps

The Allen-Sherman-Hoff Company of Philadelphia has just issued a new catalog printed in color, illustrating and describing its line of "Hydroseal," rubber-lined pumps for handling fluids containing abrasive materials such as sand, slime, sludge and slurry. The hydroseal principle is explained and the characteristics of the pumps are given as well as specification tables.

Materials-Handling Machinery

This 112-page catalog of the Gifford-Wood Company, Hudson, N. Y., covers that company's complete line of materials-handling machinery of which a considerable portion deals with boiler house equipment such as conveyors, feeders, elevators, gates, bunkers, drag scrapers, buckets, silos, weigh larries, etc. The various types are described and much useful data are included.

Measuring Instruments

Of interest to power plant executives, engineers and operating men, whether connected with central stations or with industrial plants, is a catalog-type broadside No. 160 on "Power Plant Measuring Instruments, Telemeters and Automatic Controls," just issued by Leeds & Northrup Company, Philadelphia. Equipments for many power plant measurements and controls are shown and described in compact yet remarkably complete form. Specific applications are mentioned—in electrical generation and transmission, in steam generation and distribution, in hydropower generation and in diesel power generation—in which these equipments are being used to safeguard operation and to effect operating economies.

Monel Metal

A new booklet describing the solution to scores of actual metal problems as encountered by the engineer has just been issued by The International Nickel Company. Containing 48 pages, illustrated throughout, this booklet has been prepared primarily as a guide book to Monel and other non-ferrous nickel alloys in the fields of engineering applications. It also covers the corrosion resistance and other properties of these metals, and has twenty sub-divisions, each devoted to specific problems in various fields including hydro-electric and steam power plants. Special items included are those dealing with meters and regulating equipment, turbine blading, valve fittings, condensers and pump maintenance.

Small Stokers

Recently off the press is a 28-page booklet, No. 1619, issued by the Link Belt Company, Chicago, describing its line of automatic stokers for industrial applications in capacities up to 300 hp. The book is fully illustrated and contains much pertinent data. Among the major headings are: Scientific combustion cycle, burning heads tailored to fit the coal available, variable intermittent drive, automatic air control, hopper design and analysis of a stoker efficiency test.

Thermometers

Data Book No. 210, issued by Republic Flow Meters Co., Chicago, features a new gas-filled recording thermomenter containing a helical coil which is cartridge-sealed as a protection against dirt, moisture and damage by handling. The helix, capillary tubing and sensitive bulb are welded together, filled with nitrogen gas, calibrated and sealed in the laboratory, to form a complete thermal unit which can be removed from the reading instrument for recalibration, repair or change of temperature range, without disturbing or exposing any mechanism. The reading instrument may be placed at any point up to 200 ft from the point of temperature measurement, with the instrument so compensated that the effects of temperature changes on the capillary may be disregarded.

Variable-Speed Control

"Speed Control at Work" is the title of a booklet just issued by Reeves Pulley Company, Columbus, O. This contains a brief inspection, both by word and illustration, of the methods employed by twelve nationally-known manufacturers in securing complete speed adjustability of production machines in their plants. These methods, and their applications, are typical of those found in thousands of industrial plants, large and small, and may profitably be studied by manufacturers interested in more efficient machine operations through variable speed control. In addition, the booklet illustrates and describes the three basic units in the line of REEVES speed control equipment and a number of the different designs and controls available.

Steam Turbine Regulation and Supervisory Control

An abstract of two papers presented at the Annual Meeting of the A. S. M. E. in which are discussed some of the regulating problems involved in the operation of superposed high-pressure turbines and also recently developed supervisory control instruments for the detection and measurement of shaft eccentricity, bearing vibration, shell expansion and interference or rubbing of rotating parts.

REGULATING problems involved in the operation of superposed turbines and turbine supervisory instruments formed the subjects of two papers at the recent A. S. M. E. Annual Meeting. The first of these subjects was discussed by A. F. Schwendner and A. A. Luoma of the Westinghouse Electric & Manufacturing Company, and the second subject by J. L. Roberts and C. D. Greentree of the General Electric Company.

According to the authors of the first paper, the superposed back-pressure turbine presents problems of control which are more difficult than those encountered so far for condensing turbines. The heat drops are usually low, leading to enormous steam flows even for moderate capacities. Since there is no limitation imposed in connection with exhaust area, 3600-rpm turbines can be employed up to the limit of the generator capacity. This leads to an abnormally small spindle inertia when compared to condensing turbines.

The superposed turbine may run as an individual unit exhausting to the low-pressure boiler header, in which case the steam flow is determined by the plant load and is controlled by the governor speed changer. In another case, the superposed turbine may supply steam only to one or several low-pressure turbines. Here the governor will have to respond to the varying flow demand of the low-pressure units and keep the pressure in the low-pressure header within close limits. The load on the superposed turbine will be determined by the steam demand of the low-pressure units and will be controlled by a speed governor which will also respond to the exhaust pressure chambers.

Finally, there is the combination where the superposed turbine is electrically tied to one or more low-pressure turbines. In that case, the governing valves of the low-pressure turbine are wide open and the combination acts as one unit with the superposed turbine governor

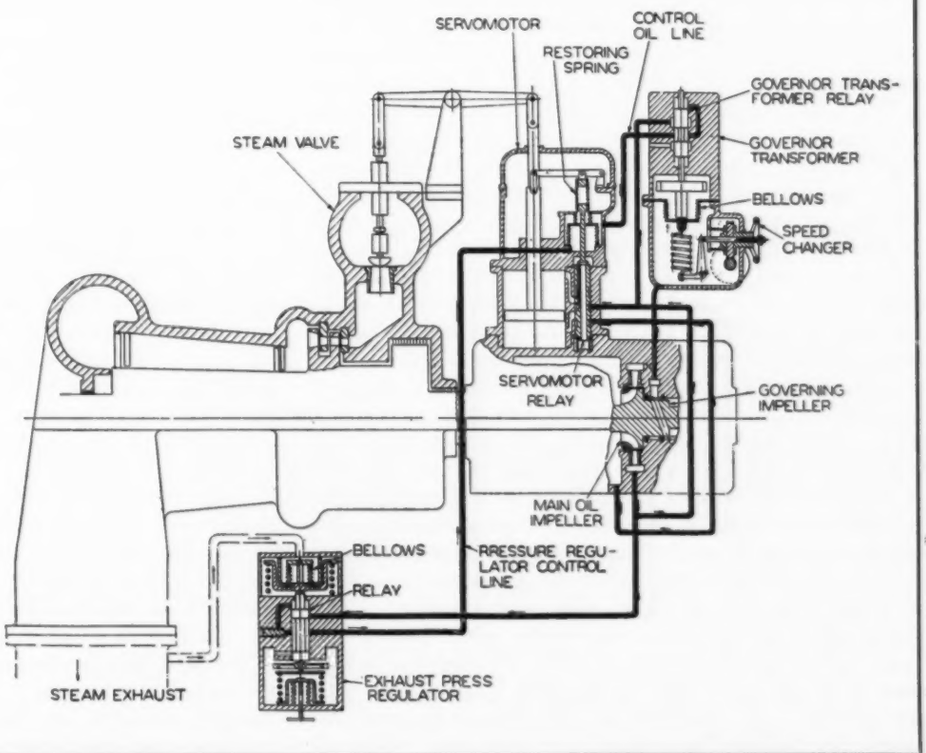


Fig. 1—Diagram of Westinghouse hydraulic governing system for steam turbines

controlling load and speed set by the governor speed changer. Through the electrical tie, the inertia of the low-pressure turbine spindle is more or less added to the superposed turbine spindle. Thus, the combination will have the governing characteristics of a condensing unit.

Stability can be obtained by proper relation between the governor-speed variation, its rapidity of response, inertia of the turbine spindle and generator rotor, governing valve sizes and other factors. The governor and connecting linkage must also be free to a considerable degree from friction and back lash.

The remainder of the paper was devoted to a description of the hydraulic governing system used on Westinghouse superposed turbines and to a theoretical analysis of synchronizing stability and stability on the line with an exhaust pressure regulator in action.

Description of Westinghouse Hydraulic Governor

In the hydraulic governing system, instead of transforming speed into motion through the medium of a flyball governor, the shaft speed is measured by the pressure in a chamber surrounding the shaft. This pressure is maintained by the centrifugal force of the oil in the inclined hole drilled in the turbine shaft connecting the pressure chamber to the drain, as shown in Fig. 1. Oil is supplied through an orifice into the pressure chamber from the main oil pump located on the turbine shaft next to the inclined hole. The small amount of oil flowing through the orifice into the pressure chamber maintains a pressure which varies as the square of the turbine speed. Excess oil supplied through the orifice above that required to maintain the pressure in this chamber passes beyond the center of the shaft and then out to drain.

To increase the sensitivity and response of the servomotor to very small speed changes, the originally small pressure change of the governor impeller is magnified in the governor transformer. This consists of a bellows which is opposed by the speed-changer spring and the pressure in the small control chamber on top of the transformer relay. The pressure in this control chamber depends on the pressure change of the governor impeller and the difference in the areas of the bellows and transformer relay. The initial oil pressure is carried by the speed-changer spring. The control chamber is connected through drilled holes to the regulating annulus located at the middle of the transformer relay. This, in turn, may communicate with high-pressure oil above and with the drain connection below, depending on the motion of the transformer relay. The transformer relay is kept revolving by a jet of oil directed against a turbine element mounted on the transformer relay. The regulating pressure is admitted to the bellows chamber of the governor servomotor. It acts on the effective area of the bellows in a downward direction and is opposed by a tension spring between the bellows and return link. The main relay is moved downward by the oil pressure above the relay, which is supplied through an orifice in the relay bushing. The compression spring below the main relay moves the relay in an upward direction. The movements of the main relay are controlled by the pilot relay attached to the bellows. An increase in the regulating pressure moves the bellows in a downward direction. The oil pressure above the main relay will increase due to the decreased flow between the main and pilot relay. The increased oil pressure will move the relay downward. The main relay in this position admits high-pressure oil under the operating piston and allows the oil on top of the piston to discharge.

The resultant upward motion of the operating piston is checked only when the return link has increased the spring tension by the same amount that the regulating pressure forced the bellows downward. This will bring the bellows, pilot and main relays into neutral position. Tracing through a change in speed, the system operates as follows: An increase in speed produces an increase in governor-impeller pressure which, in turn, increases the regulating pressure. The increased regulating pressure moves the operating piston upward which, in turn, closes the steam-admission valves.

To make the governor respond to an impulse other than speed, control pressure regulated by an exhaust-pressure regulator can be admitted below the bellows of the governor servomotor. A decrease in control pressure underneath the bellows will have the same effect as an increase in the regulating pressure above the bellows. As an increase in the exhaust pressure requires closing of the governing valves, the exhaust-pressure regulator has to be so constructed that an increase in the exhaust pressure will produce a decreased control pressure.

The exhaust pressure regulator has a bellows chamber which is connected to the turbine exhaust. The control chamber of the regulator relay is located at the top and the control pressure acts on the regulator relay in the same direction as the exhaust pressure. With the regulator relay in its neutral position, the force of the exhaust pressure on the bellows, plus the force of the oil pressure on the regulator relay, must balance the spring forces at the bellows and below the regulator relay. Since the spring force is constant at the regulator relay neutral point, an increase in exhaust pressure will decrease the control pressure. The effective areas of the bellows and regulator relay are so selected that the required exhaust-pressure change will create a control-pressure change which can move the operating piston from full-open to closed valve positions. The regulator relay is also kept revolving in the same manner as the transformer relay.

Supervisory Instruments

In the starting, stopping and normal running of turbine-generators, it has been necessary to rely on the accumulated experience from the operation of previous units to assure safe and successful performance. Starting cycles and normal running instructions were prepared by manufacturers for individual units, who recommended the procedure to be followed. The recommendations were based on observations of hundreds of machines in service and, in general, have provided safe but conservative technique for individual machines. Based on experience gained with individual units, operators have modified their practice. Unusual conditions both within and without the turbine, influence the manner in which the turbines should be operated.

The recording instruments show the presence of these unusual conditions and will enable the operator to make changes required in his operating procedure, thus avoiding, to a great extent, probability of injury to the

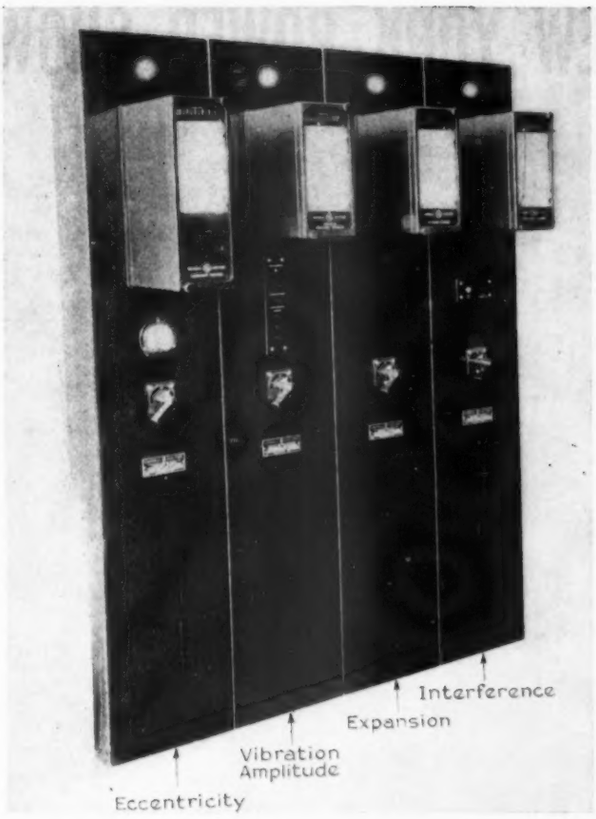


Fig. 2—Panel for turbine supervisory instruments

turbine. The four devices used are the eccentricity recorder, the vibration amplitude recorder, the expansion recorder and the interference detector. Each of these instruments uses approximately 150 watts at 115 volts, 60 cycles. The recorder charts are Telechron motor driven at a rate of 3 inches per hour, and the standard chart length is equivalent to one week's operation.

Eccentricity Recorder

The complete eccentricity recorder consists of the recording instrument and the control unit mounted on a panel shaft ring, the detector coil, mounting yoke, switch and motor-generator set—the latter to supply 500-cycle power.

The detector coil is mounted on a yoke attached to the turbine foundation, and presented to a ring on the turbine shaft. The eccentricity or wobble of the shaft induces variation in the 500-cycle current, which is fed into the control unit, whose output operates a photoelectric recorder. The recorder measures shaft eccentricity in thousandths of an inch from turning-gear speed to synchronous speed. The range of measurement is from one one-thousandth of an inch to one one-hundredth of an inch at full scale deflection. Two ranges of sensitivity are provided—one for operation at turning-gear speed and the other for all higher speeds. This is automatically controlled.

Vibration Amplitude Recorder

The vibration amplitude recorder includes the recording instrument, amplifier, time switch, detector units and a panel-mounted push-button station. A vibration detector unit is mounted on each of the turbine bearings on which the vibration is to be measured. Through the amplifier the detector output is connected to and operates the recorder. The latter is of standard type with full scale deflection equal to one one-hundredth of an inch and having one one-thousandth of an inch scale divisions. Vibrations as small as one-fourth of a mil may be measured. Bearing vibration can be recorded at any frequency from 10 to 60 cycles per second for 1800 or 3600 rpm machines.

Expansion Recorder

The complete expansion recorder consists of a recording instrument, panel-mounted, and the displacement unit. The relative axial motion between the turbine shell and the foundation operates a displacement unit placed where it is desired to measure the expansion. The remote recorder indicates expansion from cold condition on a linear scale graduated in tenths of an inch.

Interference Detector

Included in the complete interference detector is a high-gain amplifier, panel mounted, together with a loud speaker and one or more detector units. The device has a relatively flat frequency response characteristic extending into the upper range of audibility, whereby faithful reproduction of the turbine noises can be obtained. The detector units may be placed where required on the turbine, and may be connected as desired to the loud speaker which gives a more accurate indication than is possible to obtain with the usual listening rod.

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RANDOM VIEWS of the NEW YORK POWER SHOW

The official count of the attendance at the New York Power Show, held November 30 to December 5, was 39,822, or about 3000 over that of 1934. The character of the exhibits was excellent and the opinion was general that this year's Show was well worthwhile.

The Law of Tools and Equipment

By LEO T. PARKER
Attorney at Law
Cincinnati, Ohio

A review of the law, as interpreted by recent court decisions, applying to employer liability in cases of injury sustained by employees through the operation or handling of equipment. Liability under the common law is distinguished from coverage by Workmen's Compensation Laws.

IT HAS long been recognized that engineers who design plants and operate installations of equipment should have at least an elementary, but dependable, knowledge of the various circumstances under which the plant owner is liable in damages for injuries caused employees by tools and equipment.

It is well established law that all employers are duty bound, in the exercise of ordinary care, to provide a reasonably safe place for employees to work and to furnish them reasonably safe means and suitable appliances with which to execute their work, subject to the limitation that an employee assumes the ordinary risks of the employment. Also, the employers are expected to use good judgment in making an effort to safeguard employees against contraction of diseases. Yet the law is settled that where an employee is supplied with anything having open and glaring defects, or his job is apparently dangerous, and a reasonably prudent person would not undertake to do the work, then the employee is deemed contributory negligent and he is not entitled to recover damages, although, of course, he may recover compensation under state laws.

Distinction between Common and Statutory Law

The circumstances are numerous under which the owner of a plant is liable for payment of compensation under the Workmen's Compensation Laws, whereas for the same kind of injuries he would not be liable under the common law.

For example, under the common law an employer is not responsible for injuries which a workman inflicts accidentally to his fellow workman. However, the Workmen's Compensation Laws definitely specify the amount of compensation to which an injured employee is entitled, irrespective of the cause, providing the injury is sustained while the employee is acting within the scope of the employment.

On the other hand, frequently, injured employees prefer to sue for damages under the common law, rather than accept the amount of compensation fixed by the Workmen's Compensation Laws. This circumstance frequently exists where the disability results from negligence of the employer, and the employee believes that a jury will allow a verdict for a greater amount of damages than the compensation authorized by the compensation laws.

However, many states have enacted laws which clearly specify that as long as the employer complies with the State Compensation Laws, an injured employee cannot sue for damages. But, in all jurisdictions it is important to know that an employee who elects to file suit under the common law is barred from relying upon the Workmen's Compensation Laws to obtain payment, should the suit terminate adversely for him, and vice versa.

Relation of Employer and Employee

Generally speaking, the owner of a plant is not liable, either under the common law or the compensation laws, for disability sustained by any person unless the party claiming compensation proves to the satisfaction of the Court that when the injury or disease occurred, he was actually in the employ of the plant owner.

The Courts are in accord with the rule that the right of the employer to control an employee indicates the legal relation of employer and employee. In other words, the fact that an employer pays a person at regular intervals does not indicate conclusively that that person is an employee. Circumstances may exist under which a person receives payment for performing work, when in fact, he is not legally an employee. For example, a person who performs work for a stated compensation, and is permitted to perform the work in accordance with his own ideas is an "independent contractor," who is *not* entitled to compensation or damages for an injury (*165 N. Y. S. 42*).

How to Determine Liability

It is very easy for any one, who is familiar with Court decisions, to determine when the owner of a plant is liable in damages for an injury. The employer is liable only when the employee proves (1) that the thing which caused the injury was defective, or (2) that the employer knew of the defect, or was negligent in not discovering it and making the needed repairs, or (3) that the employer assured the employee that the machinery or equipment was safe, or (4) that the work was being done under the supervision of the employer and according to his instructions when the injury occurred, or (5) that the employer had notice of a defect and promised to repair it and failed to do so.

Of course, an engineer may represent the employer and the latter is responsible for any act of the engineer

which causes an injury, under the above-mentioned circumstance.

Another important point of the established law is that although the employer must use ordinary care to provide his employees with reasonably safe appliances and machinery, yet he is not duty bound to furnish the newest and most modern appliances. If the employer supplies appliances and equipment of the kind used by other ordinarily prudent plant owners, he is not negligent, and therefore not liable. Moreover, if the injury is caused by a commonplace tool, whose dangerous character is apparent, usually the employer is not liable because under these circumstances the injury results from the employee's own negligence.

For illustration, where an employer purchased a grinder and directed an employee to use it and while so doing the employee accidentally put his hand in the machine and his fingers and thumb of his right hand were cut off, the higher Court refused to hold the employer liable. Although the employee proved that the grinders in common use in other plants were safer to operate, this Court (*145 S. E. 769*) said:

"The employer does not guarantee the safety of his employees. He is *not* bound to furnish them an absolutely safe place to work in, but is required simply to use reasonable care and prudence in providing such a place. He is *not* bound to furnish the best known machinery, implements and appliances, but only such as are reasonably fit and safe and as are in general use."

It seems that the majority of engineers believe that failure to equip dangerous machinery with guards, results in the employer's positive liability for injuries to employees. However, this is not always true. Moreover, although state statutes specifically require the owners of plants to install guards, failure of the plant owner to do so does not result in his liability, if the evidence proves to the satisfaction of the Court that the injured person was injured as a direct result of his own negligence.

Still another well established law is that an employer is not expected by the law to exercise the same degree of care in supplying his employees with simple tools, as with complicated ones. In other words, since employees are bound to exercise ordinary care to protect themselves against injury, the law expects the employee to discover the defective condition of simple tools. For illustration, in one case (*298 S. W. 1017*) an employee was injured when the blade of a simple tool broke and put out his eye. The Court held the employer not liable and said:

"The Court has held that in the selection and use of simple tools, whose defects are patent and obvious and as discoverable to the servants as to the master, the servant assumes the risk of the ordinary use thereof."

Of course, if an employee used a tool knowing that it was defective, he is not entitled to recover damages for the injury. Conversely, it is true that when a workman notifies his engineer, or foreman, that equipment needs repairs, and the employer does not repair the defective equipment and directs the employee to continue his work, then the employer is liable for injuries which the employee may sustain.

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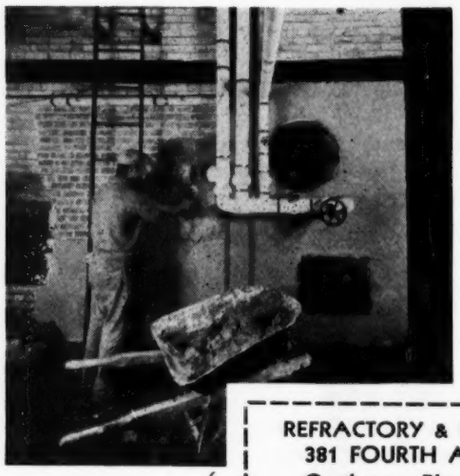

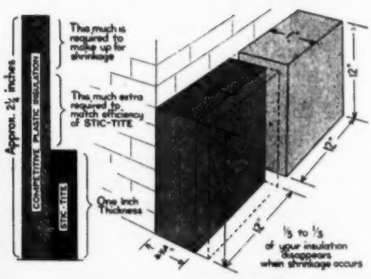
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STEAM ENGINEERING ABROAD

As reported in the foreign technical press

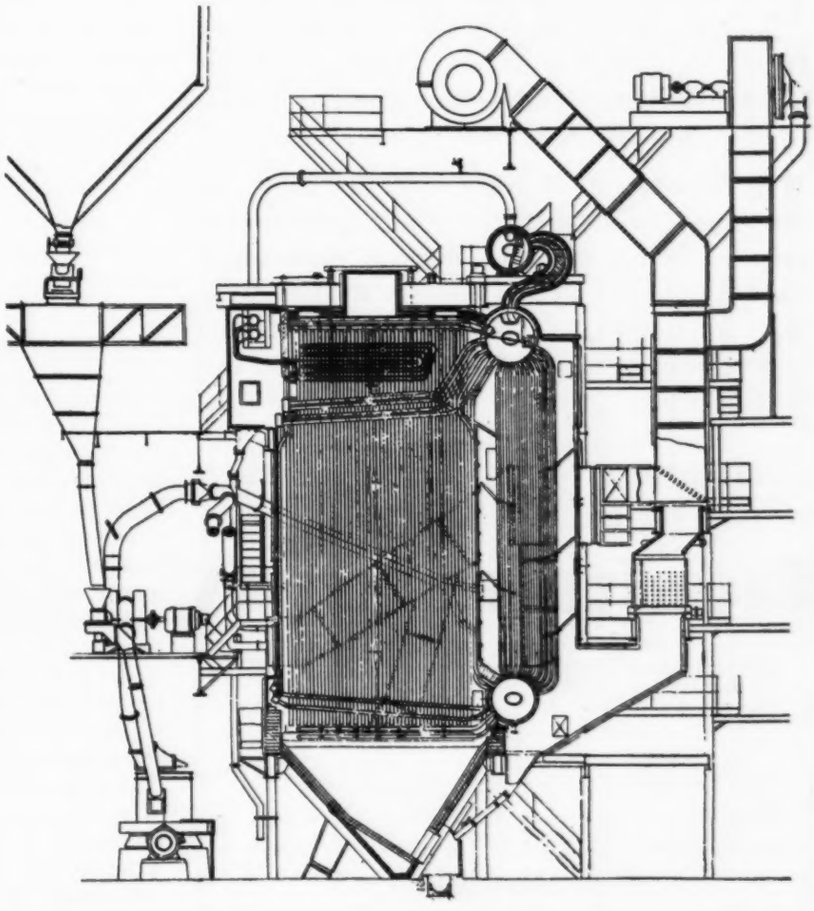
Extension at Nottingham

Four 200,000-lb per hr boilers of 625 lb pressure and 850 F steam temperature and a new 30,000-kw turbine generator constitute the latest extension to the North Wilford Power Station of the Nottingham Electricity Corporation. The new boilers are of the International Combustion Engineering steam generator type as shown in the cross-section herewith, reproduced from the October 29 issue of *The Electrical Times* (London). The Lopulco system of pulverized coal firing is employed, each mill, of the revolving-table, fixed-roller type, supplying two inter-tube burners at opposite corners of the combustion chambers. The coal contains about 15 per cent moisture and mill drying is employed.

Each unit is equipped with a Melesco superheater, economizer and air heater of the plate type designed to preheat the air to 450 F. The boiler drums are all of the hollow-forged type, the largest having 54-in. diameter with $4^3/_4$-in. wall thickness.

The turbine is of the two-cylinder impulse-reaction type with 22 stages in the high-pressure cylinder and 7 double-flow stages in the low-pressure cylinder. Three stages of bleeding for feed-water heating are incorporated and the turbine speed is 3000 rpm.

Sturtevant electrostatic dust precipitators are installed on the roof to arrest the fly ash. These precipitators consist of banks of tubes through which the gases pass, each tube having a central wire electrode charged to a high potential, the tubes being grounded. A rapping device is employed to cause the dust to fall into the hoppers. Fly ash and cinders from the hoppers, uptakes and dust collectors are then led to a water sluice beneath the boilers and pumped to fill.



Cross-section through the new steam generator at Nottingham

Detecting Cracks at Rivet Holes

This subject is discussed at length by H. Wilhelm and H. Elsasser in a recent publication of *Vereinigung der Grosskesselbesitzer* (Berlin). It appears that several methods of detecting such cracks are in use abroad. One consists of removing certain rivets, reaming and polishing the holes, and then subjecting them to visual inspection, but for this means a trained eye is required. A second method, called the Baumann method, requires polishing of the surface which is then wetted with a sulphide solution that penetrates the cracks. After drying the surface, a bromine-silver coated paper is pressed against the surface and any solution remaining in the cracks registers on the paper. This method also requires considerable experience for good results.

X-raying is also employed. This does not necessitate removal of the rivets and one exposure will cover up to twenty rivets. For plate thicknesses over $2^3/_4$ in. gamma rays are necessary for definite results. This method has one shortcoming in that the direction of the rays and

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that of the cracks frequently is not the same and reading of the film in such cases requires long experience for the detection of all cracks.

It has been found that cracks in a steel plate placed in a strong magnetic field will cause iron filings to arrange themselves along the edges of cracks. However, in order to cover all possible cracks it becomes necessary to set up two magnetic fields, normal to one another. While this procedure is reliable it has the disadvantage of attaching and moving the heavy electro magnets; also, direct current is necessary and may not always be available.

A magnetic powder method of detecting cracks has more recently come into use and seems to combine effectiveness with convenience. With this the rivets are removed and the holes reamed but polishing is not necessary. Into the rivet hole is inserted a conductor through which flows a heavy alternating current of low voltage, about 2 volts, and around this is poured mineral oil containing specially prepared and dyed iron powder. This powder will accumulate as a distinctly visible ridge over a crack. The apparatus weighs only about 55 lb, is compact and can be inserted into the drum through the manhole. Licenses for the use of this method are handled by the German Government's Materials Testing Bureau which developed the procedure.

A somewhat similar method is understood to be in use by one of the turbine builders in the United States.

Tension on Turbine Casing Bolts

In describing the new turbine-generator recently supplied by the British Thomson-Houston Company for the Pretoria (South Africa) Power Station, *The Electrical Times* mentions that a small transformer was supplied to provide a suitable electric current to heat the turbine casing bolts when assembling. This heating is done by inserting through central holes in the bolts, carbon rods connected into the transformer circuit. By controlling this heating current the correct tension on the casing bolts is obtained when they have cooled.

Air Blast Pulverizer on Shipboard

Gesellschaft für La Mont-Kessel und Kraftwirtschaft has issued reprints of an article in *Der Deutsche Seeman* by H. Pichunder, Chief Engineer of the Hamburg American Line, describing and giving operating results of the La Mont boiler installation on the steamer "Nicea."

In order to increase the speed of the vessel it was decided in 1934 to make some modification in the hull and increase the propelling power by 50 per cent. Accordingly, the two existing cylindrical boilers were replaced by two pulverized-coal-fired La Mont forced-circulation steam generators operating at 265 lb and 665 F steam temperature, a new high-pressure cylinder being fitted to the engine and the condenser being altered to handle the larger volume of exhaust steam.

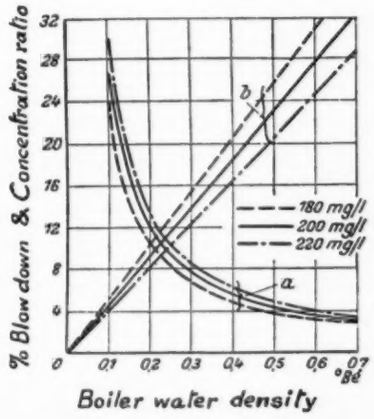
Of particular interest in this installation is the method of pulverizing the coal. This, instead of employing the usual type of mill, is effected by means of an air-blast disintegrator, called after its inventor the "Anger" disintegrator. Its design is such that the crushed coal

enters through an inclined pipe and the air, at about 800 F and a pressure of 3 lb, enters from below through a jet. This air causes the lumps of coal to impinge on an impact plate. Because of the different masses of the lumps and hence their different velocities, attrition also occurs between them. The impact plate is surrounded by a ring of adjustable vanes which act as separators and impart a spiral motion to the air-coal mixture. The primary air to the disintegrators is about one-sixth the total air required for combustion. The fineness of the coal is said to be 85 per cent through a 170-mesh screen. Regulation of the fire is accomplished by varying the speed of the primary air blowers.

The boat has now been in operation with this equipment for considerably over a year and, according to the author, no serious troubles have been encountered.

Boiler Water Concentration with Loeffler Boilers

The accompanying chart showing the blow-down concentration ratio and boiler water density for Loeffler boilers is reproduced from an abstract in the November issue of *Engineering and Boiler House Review* reporting experiences with the Loeffler boilers at the I. G. Farbenindustrie A.-G. Höchst plant (Germany). This is a superposed installation operating at about 1800-lb pressure and 930 F steam temperature. The feedwater is treated chemically and the permissible limit of boiler water density is important. The chart shows at *a* the relation between boiler water density and per cent of blowdown, and at *b* the relation between boiler water density and the ratio of concentration by evaporation. Experience has shown that operation at 0.5 Baumé is satisfactory, corresponding to a concentration ratio of 23 and 4.5 per cent blowdown, with an average feedwater salt content of 200 mg per liter. Higher density than 0.5 Baumé is considered practicable. Salt deposits in the high-pressure turbine have been practically eliminated by maintaining a low soda number.



Blowdown concentration ratio and boiler water density in Loeffler boilers at Höchst plant

The boiler plant is shut down every week-end and is started again on Monday mornings, about $1^1/_2$ hr being required to bring it up to full operating conditions.

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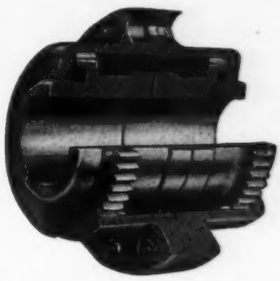

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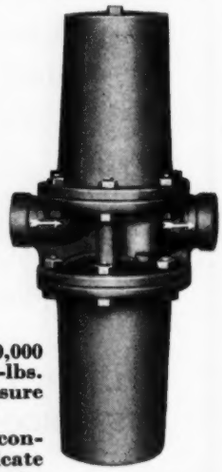

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Ross A. Langworthy has established the Ross A. Langworthy Company, 485 Madison Ave., New York, as consultants in engineering design and construction, particularly power plant work.

Leslie W. W. Morrow, for the past ten years editor of *Electrical World*, has lately joined the Corning Glass Works as manager of its new fibre products division.

F. W. Doolittle, vice president of The North American Company, retired on December 1, but will continue as a director of that company, and plans an extensive trip abroad for the study of developments in the electric utility field.

Charles F. Scott, professor emeritus of electrical engineering at Yale University, has been re-elected chairman of the Engineers' Council for Professional Development.

Walter C. Jones, for many years with United Engineers & Constructors, Philadelphia, has lately joined the engineering staff of the Public Utility Service Corporation (formerly the Byllesby Engineering & Management Corporation). His headquarters will be at Pittsburgh.

George A. Orrok, well-known consulting engineer, New York, was awarded Honorary Membership in the American Society of Mechanical Engineers at the Annual Meeting of the Society on December 1.

J. M. Pilcher has been appointed research engineer in the fuels division of Battelle Memorial Institute, Columbus, O.

Murray G. Day, for many years identified with the utility industry, and a former member of the Prime Movers Committee, has joined the staff of the Hagan Corporation, Pittsburgh.

C. A. Reed, formerly technical engineer with the Pittsburgh Coal Company, has lately joined the National Coal Association, Washington, D. C., as director of the engineering department.

W. E. Kimball, plant engineer of the South Boston refinery of the American Sugar Refining Company, died suddenly of a heart attack on December 1 while attending the Annual Meeting of the American Society of Mechanical Engineers.

Lee B. Mettler, president of the Lee B. Mettler Company, combustion engineers, Los Angeles, Calif., died of a heart attack on November 13.

Paul H. Smith, for the past ten years manager of the Detroit district of Combustion Engineering Company, Inc., died at Detroit on November 23, following a lingering illness. Mr. Smith, after graduation from Lehigh University in 1902, spent a number of years with the Westinghouse Electric & Manufacturing Company, later joining the Brooklyn Edison Company and since 1926 was with Combustion Engineering Company.

James A. Hall, professor of mechanical engineering at Brown University, Providence, R. I., and recently elected vice president of the American Society of Mechanical Engineers, died suddenly on October 29, of thrombosis, at the age of forty-eight.